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Applicant: Wataru ISHIMARU
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Sir:

Further to Applicants' claim of priority under 35 U.S.C. 119 from foreign application, Japanese Patent Application No 2002-207242, filed July 16, 2002, Applicants submit herewith an English translation of said original foreign application, and a certificate from the translator of the accuracy of the translation.

Respectfully submitted,

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Signed, this 25th day of January, 2005

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[Document Name] Specification

[Title of the Invention] Gear Change-speed Unit for Automatic transmission

[Claims]

[Claim 1] A gear change-speed unit for an automatic transmission,

5 including

an input part to which rotation is input from a power source,

an output part arranged coaxially with the input part,

three planetary-gear sets which can provide a number of transfer
paths between the input and output parts, and

10 three clutches and two brakes which can be engaged and released
selectively so that the planetary-gear sets can select one of the transfer
paths to change rotation out of the input part at a corresponding gear ratio
and provide it to the output part,

wherein at least 6 forward speeds and 1 reverse can be selected by
15 a combination of engagement and release of the clutches and brakes,
characterized in that

one of the three planetary-gear sets includes a
reduction planetary-gear set for reducing input rotation at all times for
outputting,

20 that one of the remaining two planetary-gear sets includes a double
sun-gear type planetary-gear set comprising two sun gears, a common
pinion meshed with the two sun gears, a ring gear meshed with the pinion,
and a carrier which can input and output rotation between the two sun gears
through a center member coupled to a side member for rotatably supporting
25 the pinion, and

that another planetary-gear set includes a single-pinion type
planetary-gear set comprising a sun gear, a pinion meshed with the sun
gear, a ring gear meshed with the pinion, and a carrier for rotatably
supporting the pinion,

30 wherein the three planetary-gear sets are disposed in parallel in
order of the reduction planetary-gear set, single-pinion type planetary-gear
set, and double sun-gear type planetary-gear set from the side of the input

part.

[Claim 2] The gear change-speed unit for an automatic transmission as described in claim 1, **characterized in that** the first and second clutches of the three clutches for distributing output rotation of the reduction planetary-gear set to a change-speed planetary-gear set comprising the double sun-gear type planetary-gear set and single-pinion type planetary-gear set are arranged closer to the single-pinion type planetary-gear set than the double sun-gear type planetary-gear set.

[Claim 3] The gear change-speed unit for an automatic transmission as described in claim 2, **characterized in that** clutch pistons of the first and second clutches are arranged on the side of the single-pinion type planetary-gear set distant from the double sun-gear type planetary-gear set.

[Claim 4] The gear change-speed unit for an automatic transmission as described in claim 3, **characterized in that** the third clutch of the three clutches for directly providing rotation of the input part to the change-speed planetary-gear set comprising the double sun-gear type planetary-gear set and single-pinion type planetary-gear set is arranged at the outer periphery of the reduction planetary-gear set.

[Claim 5] The gear change-speed unit for an automatic transmission as described in claim 4, **characterized in that** a clutch piston of the third clutch is arranged on the side of the reduction planetary-gear set close to the single-pinion type planetary-gear set.

[Claim 6] The gear change-speed unit for an automatic transmission as described in any of claims 1 to 5, **characterized in that** an output gear of the transmission is disposed between the reduction planetary-gear set and the single-pinion type planetary-gear set to rotatably be supported on a transmission casing, wherein hydraulic passages of the first and second clutches are formed through a wall of the output gear arranged on the transmission casing for supporting.

[Claim 7] The gear change-speed unit for an automatic transmission as described in any of claims 1 to 6, **characterized in that** the two brakes include the first and second brakes which can fix rotary members of the

change-speed planetary-gear set comprising the double sun-gear type planetary-gear set and single-pinion type planetary-gear set, wherein the brakes are arranged closer to the single-pinion type planetary-gear set than the double sun-gear type planetary-gear set.

5 [Claim 8] The gear change-speed unit for an automatic transmission as described in claim 7, **characterized in that** the first and second brakes are arranged at the outer peripheries of the first and second clutches.

[Claim 9] The gear change-speed unit for an automatic transmission as described in claim 7 or 8, **characterized in that** the first brake includes a
10 brake for fixing the carrier of the double sun-gear type planetary-gear set, and the second brake includes a brake for fixing the sun gear of the double sun-gear type planetary-gear set distant from the single-pinion type planetary-gear set, wherein the first brake is disposed closer to the reduction planetary-gear set than the second brake.

15 [Claim 10] The gear change-speed unit for an automatic transmission as described in any of claims 1 to 9, **characterized in that**

the reduction planetary-gear set which is the closest of the three planetary-gear sets to the input part is constructed by a planetary-gear set comprising a first sun gear, a first ring gear, and a first
20 carrier supporting a pinion meshed with the gears,

that the single-pinion type planetary-gear set positioned close to the input part is constructed by a planetary-gear set comprising a second sun gear, a second ring gear, and a second carrier supporting a pinion meshed with the gears, and

25 that the double sun-gear type planetary-gear set positioned the most distant from the input part includes a planetary-gear set comprising a third sun gear on the side close to the input part, a fourth sun gear on the side distant therefrom, a third carrier supporting a common pinion meshed with the sun gears, and a single third ring gear meshed with the common pinion,

30 wherein the input part is coupled to the first ring gear, and the output part is coupled to a mutual coupling unit of the second carrier and the third ring gear,

wherein there are provided a first clutch which can engage and release the first carrier from the second ring gear, a second clutch which can engage and release the first carrier from the mutual coupling unit of the second sun gear and the third sun gear, a third clutch which can engage and release the third carrier from the input part, a first brake which can fix the third carrier, and a second brake which can fix the fourth sun gear, and

wherein it is constructed to be selectable first speed by engaging the first clutch and the first brake, second speed by engaging the first clutch and the second brake, third speed by engaging the first and second clutches, fourth speed by engaging the first and third clutches, fifth speed by engaging the second and third clutches, sixth speed by engaging the third clutch and the second brake, and reverse by engaging the second clutch and the first brake.

[Claim 11] The gear change-speed unit for an automatic transmission as described in any of claims 1 to 9, **characterized in that**

the reduction planetary-gear set which is the closest of the three planetary-gear sets to the input part is constructed by a planetary-gear set comprising a first sun gear, a first ring gear, and a first carrier supporting a pinion meshed with the gears,

that the single-pinion type planetary-gear set positioned close to the input part is constructed by a planetary-gear set comprising a second sun gear, a second ring gear, and a second carrier supporting a pinion meshed with the gears, and

that the double sun-gear type planetary-gear set positioned the most distant from the input part includes a planetary-gear set comprising a third sun gear on the side close to the input part, a fourth sun gear on the side distant therefrom, a third carrier supporting a common pinion meshed with the sun gears, and a single third ring gear meshed with the common pinion,

wherein the input part is coupled to the first carrier, and the output part is coupled to a mutual coupling unit of the second carrier and the third ring gear,

wherein there are provided a first clutch which can engage and

release the first ring gear from the second ring gear, a second clutch which can engage and release the first ring gear from the mutual coupling unit of the second sun gear and the third sun gear, a third clutch which can engage and release the third carrier from the input part, a first brake which can fix the third carrier, and a second brake which can fix the fourth sun gear, and

wherein it is constructed to be selectable first speed by engaging the first clutch and the first brake, second speed by engaging the first clutch and the second brake, third speed by engaging the first and second clutches, fourth speed by engaging the first and third clutches, fifth speed by engaging the second and third clutches, sixth speed by engaging the third clutch and the second brake, and reverse by engaging the second clutch and the first brake.

[Claim 12] The gear change-speed unit for an automatic transmission as described in claim 10 or 11, **characterized in that**

the output part is disposed between the reduction planetary-gear set and the single-pinion type planetary-gear set, and is coupled to a mutual coupling unit of the second carrier and the third ring gear through a tubular coupling member,

that the first and second clutches are arranged at the inner periphery of the tubular coupling member, and the first and second brakes are arranged at the outer periphery of the tubular coupling member,

that the first brake is disposed closer to the input part than the second brake, and is coupled to an outer member extending radially outward from the third carrier in a roughly middle position of a pinion axis of the double sun-gear type planetary-gear set,

that the second brake is coupled to the fourth sun gear through a radial member extending radially outward from the fourth sun gear, and

that the third clutch is disposed at the outer periphery of the reduction planetary-gear set, and has a clutch drum coupled to the third carrier through an intermediate shaft arranged through the center of the single-pinion type planetary-gear set and the double sun-gear type planetary-gear set and the center member extending radially inward from

the third carrier via between the third and fourth sun gears.

[Claim 13] The gear change-speed unit for an automatic transmission as described in any of claims 10 to 12, **characterized in that** the reduction planetary-gear set includes single-pinion type planetary-gear set.

5 [Claim 14] The gear change-speed unit for an automatic transmission as described in any of claims 10 to 12, **characterized in that** the reduction planetary-gear set includes double-pinion type planetary-gear set.

[Detailed Description of the Invention]

[0001]

10 [Technical Field to which the Invention Pertains]

The present invention relates to a gear change-speed unit for an automatic transmission, constructed by an input part, three planetary-gear sets, three clutches, two brakes, and an output part, wherein at least 6 forward speeds and 1 reverse are obtained by engaging/releasing as appropriate the three clutches and two brakes serving as change-speed elements.

[0002]

[Prior Art]

It is conventionally proposed, as shown, for example, in FIG. 7 of Japanese Published Unexamined Patent Application 4-219553, a gear change-speed unit for an automatic transmission, constructed by an input shaft, a single-pinion type planetary-gear set, a Simpson-type planetary-gear train including a combination of two single-pinion type planetary-gear sets, three clutches, two brakes, and an output shaft, wherein 6 forward speeds and 1 reverse are obtained by engaging/releasing as appropriate the three clutches and two brakes serving as change-speed elements.

[0003]

The gear change-speed unit constructed by a single-pinion type planetary-gear set and Simpson-type planetary-gear train in such a way has the features enumerated below:

(1) It is advantageous in strength, since torque-transfer flow at first

speed where torque becomes maximum in the Simpson-type planetary-gear train is shared among all members, and

(2) It is advantageous in gear strength and gear life, carrier rigidity, and the like, since the Simpson-type planetary-gear train adopts ring-gear input, wherein tangential force is about half as compared with sun-gear input.

[0004]

[Problem to be Solved by the Invention]

Incidentally, as described above, the conventional gear change-speed unit comprising a single-pinion type planetary-gear set and Simpson-type planetary-gear train had the above advantages, but raised the following problems:

(3) Carrier input to the Simpson-type planetary-gear train is needed to obtain overdrive (O/D) speed. If input and output shafts are arranged coaxially, an input path to the carrier is not established in the single-pinion type planetary-gear set wherein the number of rotary members is limited to three, and

(4) Therefore, the necessity of arranging the input and output shafts on different axes in parallel-axis layout occurs to establish an input path to the carrier, resulting in an increase in radial size of the automatic transmission.

[0005]

In order to solve the above problems (3) and (4), Japanese Published Unexamined Patent Application 4-219553 also proposes, in FIGS. 13, 14 and 15, a gear change-speed unit using a Ravigneaux-type compound planetary-gear train in place of Simpson-type planetary-gear train.

[0006]

However, the change-speed unit adopting Ravigneaux-type compound planetary-gear train allows coaxial arrangement of all components to avoid parallel-axes arrangement, but presents the problems listed below:

(5) It is disadvantageous in strength, since maximum torque of the gear train (at first speed) is borne by one double-pinion type planetary-gear set of the Ravigneaux-type compound planetary-gear train,

5 (6) It is disadvantageous in gear strength, gear life, carrier rigidity, and the like, since torque increased by one single-pinion type planetary-gear set as a reduction gear is input to a sun gear of the Ravigneaux-type compound planetary-gear train, which increases a tangential force as compared with ring-gear input,

10 (7) It is necessary to enlarge the Ravigneaux-type compound planetary-gear train, as a result, the automatic transmission, since both achievement of the strength (mechanical strength and gear life) of the Ravigneaux-type compound planetary-gear train at first speed and enhancement in carrier rigidity and the like are required, and

15 (8) Fuel consumption is degraded, since the Ravigneaux-type compound planetary-gear train has torque circulation occurring at certain speeds to reduce the transfer efficiency thereat.

[0007]

In other words, as described above, the gear change-speed unit including a combination of one single-pinion type planetary-gear set and
20 Ravigneaux-type compound planetary-gear train loses both of the above advantages (1) and (2) which are strengths of the gear change-speed unit including a combination of one single-pinion type planetary-gear set and Simpson-type planetary-gear train. As for enlargement discussed at the above (3) and (4), for another reason that the Ravigneaux-type compound
25 planetary-gear train is enlarged, enlargement of the automatic transmission cannot be avoided consequently, thus failing to solve the problem of enlargement.

[0008]

In view of the above actual situations, the present invention has its
30 main object the provision of a gear change-speed unit for an automatic transmission, which can solve, by achieving coaxial arrangement of the input and output parts without using the Ravigneaux-type compound

planetary-gear train, not only the problem of enlargement discussed at the above (3) and (4), but also the problem discussed at (8) concerning torque circulation when using the Ravigneaux-type compound planetary-gear train while maintaining the advantages discussed at the above (1) and (2) when using the Simpson-type planetary-gear train, i.e. preserving the strength advantage (gear strength, gear life, and the like) of the gear train, thus allowing degradation of fuel consumption to be avoided and gear-ratio selection flexibility to be enhanced as compared with when using the Ravigneaux-type planetary-gear train.

10 [0009]

Further, the present invention has its object the provision of a gear change-speed unit for an automatic transmission having not only the above advantages, but also excellent vehicle mountability by allowing, when mounting the above gear change-speed unit in an automotive engine room in horizontal disposition, the outer periphery of an end distant from the input part to have a small diameter so as not to interfere with the vehicle member protruding in the engine room.

[0010]

[Means for Solving the Problem]

20 For that purpose, the gear change-speed unit for an automatic transmission according to the present invention has as its precondition, as described in claim 1, a gear change-speed unit for an automatic transmission including

an input part to which rotation is input from a power source,
25 an output part arranged coaxially with the input part,
three planetary-gear sets which can provide a number of transfer paths between the input and output parts, and

three clutches and two brakes which can be engaged and released selectively so that the planetary-gear sets can select one of the transfer paths to change rotation out of the input part at a corresponding gear ratio
30 and provide it to the output part,

wherein at least 6 forward speeds and 1 reverse can be selected by

a combination of engagement and release of the clutches and brakes.

[0011]

In the present invention, one of the three planetary-gear sets includes a reduction planetary-gear set for reducing input rotation at all
5 times for outputting.

And one of the remaining two planetary-gear sets includes a double sun-gear type planetary-gear set comprising two sun gears, a common pinion meshed with the two sun gears, a ring gear meshed with the pinion, and a carrier which can input and output rotation between the two sun gears
10 through a center member coupled to a side member for rotatably supporting the pinion, and

another planetary-gear set includes a single-pinion type planetary-gear set comprising a sun gear, a pinion meshed with the sun gear, a ring gear meshed with the pinion, and a carrier for rotatably
15 supporting the pinion.

[0012]

Further, in the present invention, the three planetary-gear sets are disposed in parallel in order of the reduction planetary-gear set, single-pinion type planetary-gear set, and double sun-gear type
20 planetary-gear set from the side of the input part.

[0013]

[Effect of the Invention]

According to the gear change-speed unit of the present invention, since it comprises a combination of the reduction planetary-gear set, single-pinion type planetary-gear set, double sun-gear type planetary-gear
25 set, it is advantageous in strength, since torque-transfer flow from the reduction planetary-gear set at first speed where torque becomes maximum is carried out through all members of the Simpson-type planetary-gear set and the double sun-gear type planetary-gear set. Additionally, it is
30 advantageous in gear strength and life and carrier rigidity, since the rotary member to which torque is input from the reduction planetary-gear set is not the sun gear of the single-pinion type planetary-gear set and double

sun-gear type planetary-gear set to achieve ring-gear input or carrier input, reducing by half a tangential force.

That is, the advantages discussed at the above (1) and (2) when using the Simpson-type planetary-gear set can be maintained.

5 [0014]

Further, according to the gear change-speed unit of the present invention, one of the two planetary-gear sets which form change-speed planetary-gear sets for carrying out change speed by torque input from the reduction planetary-gear set includes the above unique double sun-gear type planetary-gear set in which two sun gears exist, and a member for inputting and outputting rotation to the carrier of the double sun-gear type planetary-gear set includes the center member disposed between the two sun gears and coupled to a side member of the carrier for rotatably supporting the pinion. Thus, even when the need for transferring input rotation to the carrier in the change-speed planetary-gear set occurs to achieve overdrive speed, transfer of the input rotation to the carrier is possible through the center member between the two sun gears of the double sun-gear type planetary-gear set. This allows achievement of overdrive speed without setting the input and output parts in parallel-axis arrangement, i.e. with the input and output parts set in coaxial arrangement, resulting in cancellation of the above problems (3) and (4) about radial enlargement, which were inevitable when using the Simpson-type planetary-gear set.

[0015]

25 Still further, according to the present invention, the solution of the problems is achieved by the double sun-gear type planetary-gear set without relying on the Ravigneaux-type compound planetary-gear train. While when using the Ravigneaux-type compound planetary-gear train, the strength problem arises, i.e. in this case, the strength disadvantage is produced by carrying maximum torque of the gear train (at first speed) by one double-pinion type planetary-gear set of the Ravigneaux-type compound planetary-gear train only, the above problems (3) and (4) can be

30

resolved without involving such detriment as is apparent from the foregoing.

[0016]

Furthermore, when using the Ravigneaux-type compound planetary-gear train, since torque increased by the reduction planetary-gear set is input to the sun gear of the Ravigneaux-type compound planetary-gear train, a tangential force is enlarged as compared with ring-gear input and carrier input, having the disadvantage in terms of gear strength and life, carrier rigidity, and the like. On the other hand, according to the solution of the present invention using the double sun-gear type planetary-gear set, the above problems (3) and (4) can be resolved without involving the above disadvantage as is apparent from the foregoing.

[0017]

Further, when using the Ravigneaux-type compound planetary-gear train, torque circulation occurs in the Ravigneaux-type compound planetary-gear train at certain speeds, fuel consumption is degraded at speed at which torque circulation occurs due to a reduction in transfer efficiency. On the other hand, according to the gear change-speed unit using the double sun-gear type planetary-gear set, no torque circulation occurs to allow degradation of fuel consumption to be avoided.

Additionally, according to the gear change-speed unit of the present invention using the double sun-gear type planetary-gear set, the selection flexibility of the gear ratio can be enhanced as compared with when using the Ravigneaux-type compound planetary-gear train.

[0018]

Moreover, according to the present invention, since the three planetary-gear sets are disposed in parallel in order of the reduction planetary-gear set, single-pinion type planetary-gear set, and double sun-gear type planetary-gear set from the side of the input part, the following operational effect can be obtained:

Due to higher layout flexibility of the axial position of the ring gear at the outer periphery of the double sun-gear type planetary-gear set which is located at the rear end the most distant from the input part for disposition of

the above planetary-gear sets, the ring gear can be located close to the input part,

so that the transmission casing can radially be reduced in the vicinity of the rear-end outer peripheral portion of the double sun-gear type planetary-gear set without being obstructed by the ring gear.

As a result, the gear change-speed unit according to the present invention has not only the above various advantages, but also excellent vehicle mountability by allowing, when mounting the gear change-speed unit in the automotive engine room in horizontal disposition, the outer periphery of an end distant from the input part to have a small diameter so as not to interfere with the vehicle member protruding in the engine room.

[0019]

Incidentally, in the present invention, as described in claim 2, the first and second clutches of the three clutches for distributing output rotation of the reduction planetary-gear set to a change-speed planetary-gear set comprising the double sun-gear type planetary-gear set and single-pinion type planetary-gear set are preferably arranged closer to the single-pinion type planetary-gear set than the double sun-gear type planetary-gear set.

In this case, even when the first and second clutches are arranged on the side of the change-speed planetary-gear set, arrangement is such that those clutches are close to the input part with respect to the change-speed planetary-gear set. Thus, the range in which the transmission casing can radially be reduced in the vicinity of the rear-end outer peripheral portion of the double sun-gear type planetary-gear set can be broadened regardless of existence of the first and second clutches, allowing more noticeable achievement of an operational effect about vehicle mountability when the gear change-speed unit according to the present invention is horizontally disposed in the engine room.

Moreover, according to the above disposition of the first and second clutches, those are close to the reduction planetary-gear set so that the length of the member for providing coupling between the first and second

clutches and the reduction planetary-gear set can be shortened, achieving also shortening, downsizing, weight reduction, and simplification of the coupling member.

Further, due to horizontal disposition of the first and second clutches, their operating pistons can be set as a double piston, allowing size reduction thereof.

[0020]

When adopting the above disposition of the first and second clutches, as described in claim 3, clutch pistons of the first and second clutches are preferably arranged on the side of the single-pinion type planetary-gear set distant from the double sun-gear type planetary-gear set.

In this case, the pistons of the first and second clutches do not exist at the outer periphery of the double sun-gear type planetary-gear set, allowing noticeable achievement of an operational effect that the transmission casing is radially reduced in the vicinity of the rear-end outer peripheral portion of the double sun-gear type planetary-gear set to enhance vehicle mountability when horizontally disposing the gear change-speed unit in the engine room.

[0021]

Incidentally, the third clutch of the three clutches for directly providing rotation of the input part to the change-speed planetary-gear set comprising the double sun-gear type planetary-gear set and single-pinion type planetary-gear set is preferably arranged at the outer periphery of the reduction planetary-gear set as described in claim 4.

In this case, the third clutch is arranged in the position closer to the input part than the first and second clutches. As a consequence, the operational effect can be secured that the transmission casing is radially reduced in the vicinity of the rear-end outer peripheral portion of the double sun-gear type planetary-gear set to enhance vehicle mountability. Additionally, it provides easy routing of the hydraulic passages of the three clutches and smaller difference in passage length, allowing enhanced controllability of the clutches and uniformed shift response of shifting

involved in the clutches.

[0022]

A clutch piston of the third clutch arranged in such a way is preferably arranged on the side of the reduction planetary-gear set close to the single-pinion type planetary-gear set as described in claim 5.

In this case, the clutch piston of the third clutch is arranged opposite with and close to the clutch pistons of the first and second clutches arranged as in claim 3. Thus, the operational effect can further be secured that it provides easy routing of the hydraulic passages of the clutches and smaller difference in passage length, allowing enhanced controllability of the clutches and uniformed shift response of shifting involved in the clutches.

[0023]

Here, when forming the hydraulic passage of the first and second clutches, as described in claim 6, an output gear of the transmission is preferably disposed between the reduction planetary-gear set and the single-pinion type planetary-gear set to rotatably be supported on a transmission casing, wherein the hydraulic passages of the first and second clutches are formed through a wall of the output gear arranged on the transmission casing for supporting.

In this case, the hydraulic passages to be extended between the shift-control control-valve body mounted at an optional circumferential spot of the transmission casing and the first and second clutches can be shortened, and set to have roughly the same length, allowing uniformization of the shift response of the clutches.

[0024]

As described in claim 7, the two brakes include preferably the first and second brakes which can fix rotary members of the change-speed planetary-gear set comprising the double sun-gear type planetary-gear set and single-pinion type planetary-gear set, wherein the brakes are arranged closer to the single-pinion type planetary-gear set than the double sun-gear type planetary-gear set.

In this case, even if the two brakes are arranged in association with the change-speed planetary-gear set comprising the double sun-gear type planetary-gear set and single-pinion type planetary-gear set, they are disposed to be shifted in the direction of the input part, so that no inhibition is produced about the operational effect that the transmission casing is radially reduced in the vicinity of the rear-end outer peripheral portion of the double sun-gear type planetary-gear set to enhance vehicle mountability. Additionally, the lengths of the hydraulic passages of the brakes can be set to be roughly equal, allowing uniformization of the shift response of shifting involved in the brakes.

[0025]

The first and second brakes are preferably arranged at the outer peripheries of the first and second clutches as described in claim 8.

In this case, the coupling members for ensuring coupling between the rotary members to be fixed by the brakes and the brakes can be arranged on the side of the double sun-gear type planetary-gear set distant from the single-pinion type planetary-gear set, resulting in easy securing of mounting spaces of the coupling members.

[0026]

As described in claim 9, it is preferable that the first brake includes a brake for fixing the carrier of the double sun-gear type planetary-gear set, and the second brake includes a brake for fixing the sun gear of the double sun-gear type planetary-gear set distant from the single-pinion type planetary-gear set, wherein the first brake is disposed closer to the reduction planetary-gear set than the second brake.

In this case, routing of the coupling member for ensuring coupling between the carrier of the double sun-gear type planetary-gear set to be fixed by the first brake and the first brake and the coupling member for ensuring coupling between the sun gear of the double sun-gear type planetary-gear set distant from the single-pinion type planetary-gear set to be fixed by the second brake and the second brake becomes easy in connection with the positions of the carrier and sun gear, having the great

advantage in cost, rigidity, and space efficiency due to shortening of the coupling members.

[0027]

For a change-speed gear unit which can produce the
5 aforementioned various operational effects, the change-speed gear unit
having the following structure as described in claim 10 is practical:

The reduction planetary-gear set which is the closest of the three
planetary-gear sets to the input part is constructed by a planetary-gear set
comprising a first sun gear, a first ring gear, and a first carrier supporting a
10 pinion meshed with the gears,

the single-pinion type planetary-gear set positioned close to the
input part is constructed by a planetary-gear set comprising a second sun
gear, a second ring gear, and a second carrier supporting a pinion meshed
with the gears, and

15 the double sun-gear type planetary-gear set positioned the most
distant from the input part includes a planetary-gear set comprising a third
sun gear on the side close to the input part, a fourth sun gear on the side
distant therefrom, a third carrier supporting a common pinion meshed with
the sun gears, and a single third ring gear meshed with the common pinion.

20 And the input part is coupled to the first ring gear, and the output
part is coupled to a mutual coupling unit of the second carrier and the third
ring gear,

there are provided a first clutch which can engage and release the
first carrier from the second ring gear, a second clutch which can engage
25 and release the first carrier from the mutual coupling unit of the second sun
gear and the third sun gear, a third clutch which can engage and release the
third carrier from the input part, a first brake which can fix the third carrier,
and a second brake which can fix the fourth sun gear, and

it is constructed to be selectable first speed by engaging the first
30 clutch and the first brake, second speed by engaging the first clutch and the
second brake, third speed by engaging the first and second clutches, fourth
speed by engaging the first and third clutches, fifth speed by engaging the

second and third clutches, sixth speed by engaging the third clutch and the second brake, and reverse by engaging the second clutch and the first brake.

[0028]

5 For another practical change-speed gear unit which can produce the aforementioned various operational effects, the change-speed gear unit having the following structure as described in claim 11 is advantageous:

The reduction planetary-gear set which is the closest of the three planetary-gear sets to the input part is constructed by a planetary-gear set
10 comprising a first sun gear, a first ring gear, and a first carrier supporting a pinion meshed with the gears,

the single-pinion type planetary-gear set positioned close to the input part is constructed by a planetary-gear set comprising a second sun gear, a second ring gear, and a second carrier supporting a pinion meshed
15 with the gears, and

the double sun-gear type planetary-gear set positioned the most distant from the input part includes a planetary-gear set comprising a third sun gear on the side close to the input part, a fourth sun gear on the side distant therefrom, a third carrier supporting a common pinion meshed with
20 the sun gears, and a single third ring gear meshed with the common pinion.

And the input part is coupled to the first carrier, and the output part is coupled to a mutual coupling unit of the second carrier and the third ring gear,

there are provided a first clutch which can engage and release the
25 first ring gear from the second ring gear, a second clutch which can engage and release the first ring gear from the mutual coupling unit of the second sun gear and the third sun gear, a third clutch which can engage and release the third carrier from the input part, a first brake which can fix the third carrier, and a second brake which can fix the fourth sun gear, and

30 it is constructed to be selectable first speed by engaging the first clutch and the first brake, second speed by engaging the first clutch and the second brake, third speed by engaging the first and second clutches, fourth

speed by engaging the first and third clutches, fifth speed by engaging the second and third clutches, sixth speed by engaging the third clutch and the second brake, and reverse by engaging the second clutch and the first brake.

5 [0029]

With any of the practical change-speed gear units, it is preferable, as described in claim 12, that:

the output part is disposed between the reduction planetary-gear set and the single-pinion type planetary-gear set, and is coupled to a mutual
10 coupling unit of the second carrier and the third ring gear through a tubular coupling member,

the first and second clutches are arranged at the inner periphery of the tubular coupling member, and the first and second brakes are arranged at the outer periphery of the tubular coupling member,

15 the first brake is disposed closer to the input part than the second brake, and is coupled to an outer member extending radially outward from the third carrier in a roughly middle position of a pinion axis of the double sun-gear type planetary-gear set,

the second brake is coupled to the fourth sun gear through a radial
20 member extending radially outward from the fourth sun gear, and

the third clutch is disposed at the outer periphery of the reduction planetary-gear set, and has a clutch drum coupled to the third carrier through an intermediate shaft arranged through the center of the single-pinion type planetary-gear set and the double sun-gear type
25 planetary-gear set and the center member extending radially inward from the third carrier via between the third and fourth sun gears.

The change-speed gear set having such structure can produce all of the aforementioned operational effects while reducing the radial and axial dimensions.

30 [0030]

With any of the practical change-speed gear units, the reduction planetary-gear set can include single-pinion type planetary-gear set as

described in claim 13.

In this case, there is an advantage that it is advantageous not only in gear noise and transfer efficiency, but also in cost.

[0031]

5 [Mode for Carrying Out the Invention]

The modes for carrying out the present invention will be described in detail hereafter in accordance with the drawings.

(First Mode)

10 FIG. 1 schematically shows a gear change-speed unit for an automatic transmission according to a mode for carrying out the present invention, wherein G1 is a first planetary-gear set, G2 is a second planetary-gear set, G3 is a third planetary-gear set, M1 is a first coupling member, M2 is a second coupling member, C1 is a first clutch, C2 is a second clutch, C3 is a third clutch, B1 is a first brake, B2 is a second brake, 15 Input is an input part (input shaft 1), and Output is an output part (output gear 2).

[0032]

The gear change-speed unit for an automatic transmission (referred to as a reduction single-pinion type) according to the present mode has first 20 planetary-gear set G1 as reduction gear comprising a single-pinion type planetary-gear set, second planetary-gear set G2 of the single-pinion type, and third planetary-gear set G3 of the double-sun-gear type arranged coaxially in order from a left end (end close to the input part Input in FIG. 1.

25 A reduction planetary-gear set is constructed by the first planetary-gear set G1, whereas a change-speed planetary-gear set (refer hereafter to as Ishimaru-type planetary-gear train occasionally) is constructed by the second planetary-gear set G2 and the third planetary-gear set G3.

30 [0033]

The first planetary-gear set G1 includes a single-pinion type planetary-gear set (reduction planetary-gear set) having a first sun gear S1,

a first ring gear R1, and a first carrier PC1 for rotatably supporting a first pinion P1 meshed with the gears S1, R1.

The second planetary-gear set G2 includes a single-pinon type planetary-gear set having a second sun gear S2, a second ring gear R2, and a second carrier PC2 for rotatably supporting a second pinion P2 meshed with the gears S2, R2.

[0034]

The third planetary-gear set G3 includes a double-sun-gear type planetary-gear set having a third sun gear S3 close to the input part Input and a fourth sun gear S4 distant therefrom, a common third pinion P3 meshed with the sun gears S3, S4, a third carrier PC3 for rotatably supporting the third pinion P3, and a third ring gear R3 meshed with the third pinion P3.

The third sun gear S3 and fourth sun gear S4 are arranged coaxially, but there is no need to always set their numbers of teeth at the same.

Moreover, provided to the third carrier PC3 are a center member CM connected thereto and extending radially inward from between the sun gears S3, S4, and an outer member OM extending radially outward from the third carrier PC3. Actually, the outer member OM is put in specific disposition as will be described later.

The center member CM is integrated with the third carrier PC3, and is disposed to pass through a space which exists between adjacent third pinion P3 located on an arrangement pitch circle thereof and extend radially inward from between the sun gears S3, S4.

[0035]

The input part Input comprises input shaft 1 which is coupled to the first ring gear R1, and to an engine, not shown, as a power source through a torque converter, also not shown, so as to provide engine rotation from the input shaft 1 to the first ring gear R1.

The output part Output comprises output gear 2 which is coaxially coupled to the second coupling member M2 for providing coupling between the second carrier PC2 and the third ring gear R3 and forming an integrated

unit thereof so as to transfer transmission output rotation from the output gear 2 to automotive driving wheels through a final-gear set and a differential-gear device, not shown.

5 The first coupling member M1 is a coupling member for providing integral coupling between the second sun gear S2 and the third sun gear S3 to form an integral unit thereof.

[0036]

10 In the reduction planetary-gear set G1, the first sun gear S1 is coupled to a transmission casing 3 for fixing at all times, whereas the first carrier PC1 can be coupled as required to the second ring gear R2 through the first clutch C1 and also to the second sun gear S2 through the second clutch C2.

The center member CM of the third carrier PC3 can be coupled as required to the input shaft 1 through the third clutch C3.

15 In the double sun-gear type planetary-gear set G3, the outer member OM of the third carrier PC3 can be coupled as required to the transmission casing 3 through the first brake B1 so as to achieve fixing of the third carrier PC3 as required, whereas the fourth sun gear S4 can be coupled as required to the transmission casing 3 through the second brake B2 so as to achieve fixing of the fourth sun gear S4 as required.

20 [0037]

By engaging (indicated by circular sign) or releasing (no sign) the clutches C1, C2, C3 and the brakes B1, B2 in the combination shown in FIG. 2, the gear change-speed unit constructed as described above can select
25 corresponding speed (first to sixth forward speeds and reverse). A shift-control control-valve body (not shown) is connected to the clutches and brakes to achieve an engagement logic for this shifting.

For the shift-control control-valve body, it is adopted the hydraulically controlled type, the electronically controlled type, or the
30 combination type obtained by combining the two.

[0038]

Shift operation of the gear change-speed unit will be described in

accordance with FIGS. 2-6.

FIG. 2 shows an engagement logic of the shift elements of the gear change-speed unit as described above, FIG. 3 is an alignment chart showing the rotating state of rotary members of the gear change-speed unit at each speed, and FIGS. 4-6 are explanatory view showing torque transfer paths in the gear change-speed unit at respective speeds.

In FIG. 3, the boldest line is an alignment chart of the first planetary-gear set G1, and the bold line is an alignment chart of the change-speed planetary-gear set (Ishimaru-type planetary-gear train) comprising second planetary-gear set G2 and third planetary-gear set G3.

In FIGS. 4-6, the bold line shows a torque-transfer path of the clutches, brakes, and members, and the hatching shows gears involved in torque transfer.

[0039]

(First Speed)

As shown in FIG. 2, first forward speed is obtained by engagement of the first clutch C1 and the first brake B1.

At first speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2.

On the other hand, in the third planetary-gear set G3, since engagement of the first brake B1 causes fixing of the third carrier PC3 to the casing, rotation of the third sun gear S3 is reduced rotation having reverse direction of rotation with respect to output rotation from the third ring gear R3. Rotation of the third sun gear S3 is transferred to the second sun gear S2 of the second planetary-gear set G2 through the first coupling member M1.

[0040]

Thus, the second planetary-gear set G2 receives normal-direction reduced rotation from the second ring gear R2 and reverse-direction reduced rotation from the second sun gear S2. Rotation obtained by further decreasing reduced rotation from the second ring gear R2 is

provided from the second carrier PC2 to the output gear 2 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 3, first speed is defined by the line connecting an engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second ring gear R2 and an engagement point of the first brake B1 where rotation of the third carrier PC3 is stopped. And rotation input from the input shaft 1 is reduced and provided through the output gear 2.

[0041]

A torque-transfer path at first speed is as shown in FIG. 4(a). Torque acts on the first clutch C1, first brake B1, and members shown by the bold line and the first planetary-gear set G1, second planetary-gear set G2, and third planetary-gear set G3 (except the fourth sun gear S4) shown by the hatching.

Specifically, at first speed, the first planetary-gear set G1, second planetary-gear set G2, and third planetary-gear set G3 constituting the Ishimaru-type planetary-gear train are involved in torque transfer.

[0042]

(Second Speed)

As shown in FIG. 2, second speed can be obtained by changing of releasing the first brake B1 as engaged at first speed and engaging the second brake B2, and therefore, by engagement of the first clutch C1 and the second brake B2.

At second speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2.

On the other hand, in the third planetary-gear set G3, engagement of the second brake B2 causes fixing of the fourth sun gear S4 to the casing, thus achieving fixing of the third sun gear S3 coupled by the third pinion P3. And the second sun gear S2 coupled to the third sun gear S3 through the first coupling member M1 is fixed to the casing.

[0043]

Thus, the second planetary-gear set G2 receives normal-direction reduced rotation from the second ring gear R2, and has the second sun gear S2 fixed. Rotation obtained by further decreasing reduced rotation from the second ring gear R2 is provided from the second carrier PC2 to the output gear 2 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 3, second speed is defined by the line connecting the engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second ring gear R2 and an engagement point of the second brake B2 where rotation of the fourth sun gear S4 is stopped. And rotation input from the input shaft 1 is reduced (but higher than first speed) and provided through the output gear 2.

[0044]

A torque-transfer path at second speed is as shown in FIG. 4(b). Torque acts on the first clutch C1, second brake B1, and members shown by the bold line and the first planetary-gear set G1 and second planetary-gear set G2 shown by the hatching.

With the third planetary-gear set G3, the unconstrained third pinion P3, which simply revolves around the fixed sun gears S3, S4 with output rotation of the third ring gear R3, is not involved in torque transfer though it functions as a rotary member.

[0045]

(Third Speed)

As shown in FIG. 2, third speed can be obtained by changing of releasing the second brake B2 as engaged at second speed and engaging the second clutch C2, and therefore, by engagement of the first clutch C1 and second clutch C2.

At third speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2. Simultaneously, engagement of the second clutch C2 causes input of this reduced rotation to

the second sun gear S2 of the second planetary-gear set G2.

Thus, the second planetary-gear set G2 receives the same reduced rotation from the second ring gear R2 and the second sun gear S2. And reduced rotation (which is the same as that of the first planetary-gear set G1) is provided from the second carrier PC2 which rotates together with the gears R2, S2 to the output gear 2 through the second coupling member M2.

[0046]

That is, as shown in the alignment chart of FIG. 3, third speed is defined by the line connecting the engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second ring gear R2 and an engagement point of the second clutch C2 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second sun gear S2. And rotation input from the input shaft 1 is reduced (= reduction ratio of the first planetary-gear set G1) and provided through the output gear 2.

A torque-transfer path at third speed is as shown in FIG. 4(c). Thus, torque acts on the first clutch C1, second clutch C2, and members shown by the bold line and the first planetary-gear set G1 and second planetary-gear set G2 shown by the hatching. That is, the third planetary-gear set G3 is not involved in torque transfer at all.

[0047]

(Fourth Speed)

As shown in FIG. 2, fourth speed can be obtained by changing of releasing the second clutch C2 as engaged at third speed and engaging the third clutch C3, and therefore, by engagement of the first clutch C1 and third clutch C3.

At fourth speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2.

On the other hand, in the third planetary-gear set G3, engagement of the third clutch C3 causes input of rotation input from of the input shaft 1 to the third carrier PC3 through the center member CM.

As a result, rotation of the third sun gear S3 is increased with respect to output rotation of the third ring gear R3. This increased rotation of the third sun gear S3 is transferred to the second sun gear S2 through the first coupling member M1.

5 [0048]

Thus, the second planetary-gear set G2 receives reduced rotation from the second ring gear R2 and increased rotation from the second sun gear S2. And rotation obtained by increasing reduced rotation from the second ring gear R2 (but rotation lower than input rotation) is provided to the output gear 2 from the second carrier PC2 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 3, fourth speed is defined by the line connecting the engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second ring gear R2 and an engagement point of the third clutch C3 where rotation of the third carrier PC3 is used as input rotation. And rotation input from the input shaft 1 is slightly reduced and provided through the output gear 2.

A torque-transfer path at fourth speed is as shown in FIG. 5(a). Thus, torque acts on the first clutch C1, third clutch C3, and members shown by the bold line and the first planetary-gear set G1, second planetary-gear set G2, and third planetary-gear set G3 (except the fourth sun gear S4) shown by the hatching.

[0049]

25 (Fifth Speed)

As shown in FIG. 2, fifth speed is obtained by changing of releasing the first clutch C1 as engaged at fourth speed and engaging the second clutch C2, and therefore, by engagement of the second clutch C2 and third clutch C3.

30 At fifth speed, engagement of the second clutch C2 causes input of reduced rotation from the first planetary-gear set G1 to the third sun gear R3 through the second sun gear S2 and the first coupling member M1.

Simultaneously, engagement of the third clutch C3 causes input of input rotation from the input shaft 1 to the third carrier PC3 through the center member CM.

[0050]

5 Thus, the third planetary-gear set G3 receives input rotation at the third carrier PC3 and reduced rotation from the first planetary-gear set G1 at the third sun gear S3. Increased rotation with respect to input rotation is provided to the output gear 2 from the third ring gear R3 through the second coupling member M2.

10 That is, as shown in the alignment chart of FIG. 3, fifth speed is defined by the line connecting the engagement point of the second clutch C2 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the third sun gear S3 and the engagement point of the third clutch C3 where rotation of the third carrier PC3 is used as input rotation.
15 And rotation input from the input shaft 1 is slightly increased and provided through the output gear 2.

 A torque-transfer path at fifth speed is as shown in FIG. 5(b). Thus, torque acts on the second clutch C2, third clutch C3, and members shown by the bold line and the first planetary-gear set G1 and third
20 planetary-gear set G3 (except the fourth sun gear S4) shown by the hatching.

[0051]

(Sixth Speed)

 As shown in FIG. 2, sixth speed is obtained by changing of releasing
25 the second clutch C2 as engaged at fifth speed and engaging the second brake B2, and therefore, by engagement of the third clutch C3 and the second brake B2.

 At sixth speed, engagement of the third clutch C3 causes input of input rotation from the input shaft 1 to the third carrier PC3 through the
30 center member CM. Moreover, engagement of the second brake B2 causes fixing of the fourth sun gear S4 of the third planetary-gear set G3 to the casing.

[0052]

Thus, the third planetary-gear set G3 receives input rotation at the third carrier PC3, and has the fourth sun gear S4 fixed to the casing. Increased rotation with respect to input rotation is provided to the output gear 2 from the third ring gear R3 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 3, sixth speed is defined by the line connecting the engagement point of the third clutch C3 where rotation of the third carrier PC3 is used as input rotation and the engagement point of the second brake B2 where the fourth sun gear S4 is fixed to the casing. And rotation input from the input shaft 1 is increased and provided through the output gear 2.

A torque-transfer path at sixth speed is as shown in FIG. 5(c). Thus, torque acts on the third clutch C3, second brake B2, and members shown by the bold line and the third planetary-gear set G3 (except the third sun gear S3) shown by the hatching.

[0053]

(Reverse)

As shown in FIG. 2, reverse speed is obtained by engaging the second clutch C2 and the first brake B1.

At reverse speed, engagement of the second clutch C2 causes input of reduced rotation from the first planetary-gear set G1 to the third sun gear S3 through the second sun gear S2 and the first coupling member M1. Moreover, engagement of the first brake B1 causes fixing of the third carrier PC3 to the casing.

Thus, the third planetary-gear set G3 receives normal-direction reduced rotation at the third sun gear S3, and has the third carrier PC3 fixed to the casing. Reduced reverse rotation is provided from the third ring gear R3 to the output gear 2 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 3, reverse speed is defined by the line connecting the engagement point of the second clutch C2 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the third sun gear S3 and the engagement point of the first brake

B1 where rotation of the third carrier PC3 is stopped. And rotation input from the input shaft 1 is reduced in the reverses direction and provided through the output gear 2.

[0054]

5 A torque-transfer path at reverse is as shown in FIG. 6. Thus, torque acts on the second clutch C2, first brake B1, and members shown by the bold line and the first planetary-gear set G1 and third planetary-gear set G3 (except the fourth sun gear S4) shown by the hatching.

[0055]

10 (Advantages over the Prior Art)

The essential concept of the gear change-speed unit according to the present mode is to establish 6 forward speeds by tree clutches and two brakes using essentially a combination of the reduction planetary-gear set and the Simpson-type compound planetary-gear train with the
15 aforementioned problems (3) and (4) of the Simpson-type planetary-gear train resolved, and without producing any inevitable new problem when using the gear change-speed unit including a combination of the reduction planetary-gear set and the Ravigneaux-type compound planetary-gear train.

20 [0056]

Making a comparison with the gear change-speed unit adopting Simpson-type planetary-gear train or Ravigneaux-type compound planetary-gear train, the advantages of the gear change-speed unit according to the present mode will be described.

25 * Features of Simpson-type Planetary-gear Train

(A) It is advantageous in strength, since in the Simpson-type planetary-gear train, torque-transfer flow at first speed where maximum torque operates is shared among all members as shown in FIG. 8(a).

30 (B) It is advantageous in gear strength and life, carrier rigidity, and the like, since the Simpson-type planetary-gear train adopts ring-gear input, which reduces by half a tangential force as compared with sun-gear input. Specifically, as shown in FIG. 9, when the same torque is input to the

planetary-gear set, ring-gear input f has a tangential force reduced to $1/2-1/2.5$ as compared with sun-gear input F .

(C) Carrier input to the Simpson-type planetary-gear train is needed to obtain overdrive speed (O/D). However, if an input shaft and an output shaft are arranged coaxially, an input path to the carrier shown by the broken line in FIG. 10(b) is not established in the single-pinion type planetary-gear set, since the rotary members are limited to three members as shown in FIG. 10(a).

Thus, arrangement of the input shaft and output shaft on different parallel axes is needed to establish an input path to the carrier for actualization of overdrive speed (O/D). As a result, there arises a problem of enlarging the radial dimension of the automatic transmission.

[0057]

* Problem of Ravigneaux-type Compound Planetary-gear Train

When using the gear change-speed unit adopting Ravigneaux-type compound planetary-gear train in place of Simpson-type planetary-gear train so as to eliminate the problem of (C), coaxial arrangement of the input shaft and output shaft becomes possible, but there arises problems enumerated below:

(D) It is disadvantageous in strength, since maximum torque of the gear train (torque at first speed) is carried by one double-pinion type planetary-gear set of the Ravigneaux-type compound planetary-gear train as shown in FIG. 8(b).

(E) It is disadvantageous in gear strength and life, carrier rigidity, and the like, since torque increased by a single-pinion type planetary-gear set as reduction planetary-gear set is input to a sun gear of the Ravigneaux-type compound planetary-gear train as shown in FIG. 7, which increases a tangential force as compared with ring-gear input for the reason of (B).

(F) It is necessary to enlarge the Ravigneaux-type compound planetary-gear train, which results in enlargement of the automatic transmission, since achievement of the strength of the Ravigneaux-type

compound planetary-gear train (gear strength and life) and enhancement in carrier rigidity when first speed is selected are required.

(G) Fuel consumption is degraded, since the Ravigneaux-type compound planetary-gear train has torque circulation occurring at second speed as shown in FIG. 7 to reduce the transfer efficiency at second speed where torque circulation occurs.

Here, torque circulation is produced by divergence of output torque (2.362) and circulation torque (1.77) at the third ring gear R3. Among them, circulation torque is internally circulated through the third ring gear R3 and the second pinion P2 while second speed is selected.

[0058]

* Features of Planetary-gear Train According to the Present Mode

The features of the Ishimaru-type planetary-gear train comprising single-pinion type planetary-gear set G2 and double sun-gear type planetary-gear set G3, which is adopted in the present mode, are as follows:

(a) It is possible to arrange an input part and an output part in the same way as the Ravigneaux-type compound planetary-gear train while achieving carrier input which is indispensable for obtaining overdrive speed (O/D).

Specifically, as shown in FIG. 10(c), the double sun-gear type planetary-gear set G3 constituting Ishimaru-type planetary-gear train is larger in the number of members such that (2 members out of the sun gear) + (1 member out of the ring gear) + (2 axial and radial members out of the carrier) = 5 members, and allows, particularly, radial input between the two sun gears due to the center member. This allows carrier input which actualizes higher gear ratios (fourth to sixth speeds in the above mode) including overdrive.

(b) It is advantageous in strength, since maximum torque (first-speed transfer torque) of the gear train is carried out by both the second planetary-gear set G2 and the third planetary-gear set G3 constituting Ishimaru-type planetary-gear train as shown in FIG. 4(a) to

share first-speed torque among all members.

(c) It is advantageous in gear strength and life, carrier rigidity, and the like (downsizing is possible), since, at first and second speeds where transfer torque is larger, for example, torque increased by the first planetary-gear set G1 as reduction planetary-gear set is input from the second ring gear R2 of the Ishimaru-type planetary-gear train as shown in FIGS. 4(a) and 4(b), which allows a reduction in tangential force as compared with the Ravigneaux-type compound planetary-gear train which adopts sun-gear input.

(d) It is possible to downsize the gear change-speed unit and thus the automatic transmission, since the Ishimaru-type planetary-gear train is advantageous in strength and in gear strength and life, carrier rigidity, and the like as compared with the Ravigneaux-type compound planetary-gear train, and allows coaxial arrangement of the input part and output part in the same way as the Ravigneaux-type compound planetary-gear train.

(e) Fuel consumption is enhanced, since the Ishimaru-type planetary-gear train has no torque circulation occurring at second speed as shown in FIG. 4(b), which contributes to enhancement in transfer efficiency as compared with second speed of the Ravigneaux-type compound planetary-gear train wherein torque circulation occurs.

Specifically, FIG. 11 shows a comparison between the Ravigneaux-type compound planetary-gear train and the Ishimaru-type planetary-gear train when the gear ratio α (= sun-gear teeth number/ring-gear teeth number) is within a typically applicable range ($\alpha = 0.35-0.65$) and that consideration is made about the preferable conditions that the gear-to-gear ratio is smaller as the gear ratio is higher. As for transfer efficiency at second speed, the transfer efficiency of the Ravigneaux-type compound planetary-gear train is 0.950 or 0.952, whereas the transfer efficiency of the Ishimaru-type planetary-gear train is 0.972 when the first planetary-gear set G1 is of the single-pinion type, and 0.968 when it is of the double-pinion type.

(f) The Ravigneaux-type compound planetary-gear train has a

restriction, when setting the gear ratio α , that the number of ring-gear teeth is fixed, so that when the gear ratio is within a typically applicable range ($\alpha = 0.35-0.65$) and that consideration is made about the preferable conditions that the gear-to-gear ratio is smaller as the gear ratio is higher, the ratio coverage (= first-speed gear ratio/sixth-speed gear ratio) which is an available gear-ratio width is between 4.81 minimum and 7.20 maximum as shown in FIG. 11.

On the other hand, with the Ishimaru-type planetary-gear train wherein the gear ratios α_2, α_3 of the two planetary-gear sets G2, G3 can be set independently, the applicable ratio coverage is enlarged, as compared with the Ravigneaux-type compound planetary-gear train, to 4.81minimum-7.80maximum when the first planetary-gear set G1 is of the single-pinion type, and to 5.08minimum-9.02maximum when the first planetary-gear set G1 is of the double-pinion type as shown in FIG. 11. This results in enhanced gear-ratio selection flexibility as shown by the numerical values in FIG. 2, for example (the values of 5.5-7.0 in the uppermost column designate ratio coverage).

[0059]

As is seen from the above explanation, the following effects can be obtained with the gear change-speed unit according to the present mode:

(A) In a gear change-speed unit for an automatic transmission, including an input part Input (input shaft 1) to which rotation is input from a power source (engine), an output part Output (output gear 2) arranged coaxially with the input part, three planetary-gear sets G1, G2, G3 which can provide a number of transfer paths between the input and output parts, and three clutches C1, C2, C3 and two brakes B1, B2 which can be engaged and released selectively so that the planetary-gear sets can select one of the transfer paths to change rotation out of the input part at a corresponding gear ratio and provide it to the output part,

wherein at least 6 forward speeds and 1 reverse can be selected by a combination of engagement and release of the clutches and brakes,

since one planetary-gear set G1 of the three planetary-gear sets G1,

G2, G3 includes a reduction planetary-gear set for reducing input rotation at all times for outputting,

one planetary-gear set G3 of the remaining two planetary-gear sets includes a double sun-gear type planetary-gear set comprising two sun gears S3, S4, a common pinion P3 meshed with the two sun gears, a ring gear R3 meshed with the pinion, and a carrier PC3 which can input and output rotation between the two sun gears through a center member coupled to a side member for rotatably supporting the pinion,

another planetary-gear set G2 includes a single-pinion type planetary-gear set comprising a sun gear S2, a pinion P2 meshed with the sun gear, a ring gear R2 meshed with the pinion, and a carrier PC2 for rotatably supporting the pinion, and

a member for inputting/outputting rotation to the carrier PC3 of the double sun-gear type planetary-gear set G3 includes a center member CM disposed between the two sun gears S3, S4 and coupled to the carrier PC3,

it has the effects listed below:

(i) The strength advantage of the change-speed planetary-gear set such as gear strength and life is obtained for the reason described above, since the change-speed planetary-gear set is constructed by the two planetary-gear sets G2, G3 to serve as Ishimaru type planetary-gear train.

(ii) Fuel consumption is enhanced by removing torque circulation at second speed.

(iii) The radial dimension of the transmission can be reduced by coaxially arranging the input shaft 1 and the output gear 2.

(iv) Downsizing of the change-speed planetary-gear set is possible, since the change-speed planetary-gear set is constructed to serve as Ishimaru type planetary-gear train, which allows a reduction in strength requirements for the reason described above. This cooperates with the aforementioned coaxial arrangement of the input shaft 1 and the output gear 2 to allow downsizing of the automatic transmission.

(v) The gear-ratio selection flexibility can be enhanced for the reason described above as compared with when using Ravigneaux-type

compound planetary-gear train.

(vi) Downsizing of the reduction planetary-gear set can be achieved, since the first planetary-gear set G1 serves as reduction planetary-gear set for reducing input rotation at all times. With this, downsizing of the automatic transmission is possible.

[0060]

(B) An additional explanation will be made about each operational effect. The gear change-speed unit according to the present mode comprises a combination of the reduction planetary-gear set G1, single-pinion type planetary-gear set G2, and double sun-gear type planetary-gear set. Thus, at first speed where torque becomes maximum, torque-transfer flow of the reduction planetary-gear set G1 is carried out through all members of the single-pinion type planetary-gear set G2 and double sun-gear type planetary-gear set G3, having the strength advantage. Moreover, the rotary members for receiving torque out of the reduction planetary-gear set G1 are not sun gears of the single-pinion type planetary-gear set G2 and double sun-gear type planetary-gear set G3, and thus ring-gear input or carrier input is achieved, reducing by half a tangential force, having the advantage in gear strength and life, carrier rigidity, and the like.

That is, it is possible to maintain the aforementioned advantages (1) and (2) obtained when using the Simpson-type planetary-gear set.

[0061]

(C) Further, with the gear change-speed unit according to the present mode, one planetary-gear set G3 of the two planetary-gear sets G2, G3 constituting change-speed planetary-gear set for carrying out change speed by torque input out of the reduction planetary-gear set G1 includes a double-sun-gear type planetary-gear set in which two sun gears S3, S4 exist, and the member for inputting/outputting rotation to the carrier PC3 of the double sun-gear type planetary-gear set includes center member CM disposed between the two sun gears S3, S4 and coupled to the carrier PC3 of the double sun-gear type planetary-gear set.

Thus, even when input rotation is transferred to the carrier PC3 in the change-speed planetary-gear set for actualizing overdrive (O/D) speed, transfer of this input rotation to the carrier PC3 is possible through the center member CM disposed between the sun gears S3, S4 of the double sun-gear type planetary-gear set G3, and overdrive (O/D) speed can be actualized without putting the input part Input and output part Output in parallel-axes arrangement, i.e. with the input and output parts being arranged coaxially. This allows removal of the aforementioned problems (3) and (4) associated with a radial size increase which is inevitable when using Simpson-type planetary-gear set.

[0062]

(D) Furthermore, since the solution of the aforementioned problems is achieved by using the double sun-gear type planetary-gear set G3 without relying on the Ravigneaux-type compound planetary-gear train, the problems (3) and (4) can be eliminated without involving a strength problem raised when using the Ravigneaux-type compound planetary-gear train, i.e. strength disadvantage due to maximum torque of the gear train (at first speed) being applied to only one double-pinion type planetary-gear set of the Ravigneaux-type compound planetary-gear train.

[0063]

(E) Further, when using the Ravigneaux-type compound planetary-gear train, torque increased by the reduction planetary-gear set is input to the sun gear of the Ravigneaux-type compound planetary-gear train, having greater tangential force as compared with ring-gear input and carrier input, resulting in disadvantage in gear strength and life, carrier rigidity, and the like. On the other hand, in the solution of the present mode using the double sun-gear type planetary-gear set, the problems (3) and (4) can be eliminated without involving such disadvantage.

[0064]

(F) Still further, when using the Ravigneaux-type compound planetary-gear train, torque circulation occurs in the Ravigneaux-type compound planetary-gear train at second speed to reduce the transfer

efficiency at second speed where torque circulation occurs, thus leading to poor fuel consumption. On the other hand, with the gear change-speed unit according to the present mode using the double sun-gear type planetary-gear set G3, torque circulation does not occur to allow avoiding of poor fuel consumption.

(G) Furthermore, according to the present mode, the first planetary-gear set G1 or reduction planetary-gear set includes a single-pinion type planetary-gear set, allowing a reduction in gear noise and number of parts, leading to enhancement in the transfer efficiency and thus fuel consumption.

[0065]

FIG. 12 is an actual block diagram of the gear change-speed unit as described above. When describing hereinafter in detail the gear change-speed unit as described above in accordance with this, in FIG. 12, the input and output parts are shown in the state reversed left to right with respect to those in skeleton views in FIGS. 1 and 4-6.

The input shaft 1 and an intermediate shaft 4 are arranged in the transmission casing 3 in the butt state allowing coaxial relative rotation, the input shaft 1 and intermediate shaft 4 being supported rotatable separately with respect to the transmission casing 3.

A front-end opening of the transmission casing 3 close to the input shaft 1 is concealed by a pump casing comprising a pump housing 5 and a pump cover 6. The input shaft 1, which is arranged through the pump casing for supporting, has a protruding end drivingly coupled to an engine (not shown) or power source through a torque converter (not shown).

[0066]

A rear end of the intermediate shaft 4 distant from the input shaft 1 is rotatably supported by an end cover 7 at a rear end of the transmission casing 3.

An intermediate wall 8 is arranged in a roughly axially middle position of the transmission casing 3 to rotatably support the output gear 2. A butt engagement of the input shaft 1 and the intermediate shaft 4 is

rotatably supported in a center hole of the intermediate wall 8 through a hollow shaft 9.

[0067]

5 The first planetary-gear set G1 is disposed in a front space defined between the pump casing comprising pump housing 5 and pump cover 6 and the intermediate wall 8. And the third clutch C3 is disposed in such a way as to enclose the first planetary-gear set G1.

10 The first planetary-gear set G1 has sun gear S1 fixedly mounted to the pump cover 6 so as to disable rotation at all times, and ring gear R1 couple to a flange 10 extending radially outward from the input shaft 1.

A clutch drum 11 is arranged to extend radially outward from a front end of the intermediate shaft 4 close to the input shaft 1 so as to enclose the ring gear R1. The clutch pack 12 is arranged comprising an alternate disposition of clutch plates each splined to the inner periphery of the clutch
15 drum 11 and the outer periphery of the ring gear R1. Those members constitute third clutch C3.

A clutch piston 13 of the third clutch C3 is fitted in an end wall of the clutch drum 11 facing the first planetary-gear set G1, and makes stroke under the working oil pressure out of a hydraulic passage 14 formed
20 through the pump cover 6, the input shaft 1, and the intermediate shaft 4 so as to allow engagement of the third clutch C3.

A drum-shaped coupling member 9a is arranged to extend radially outward from a front end of the hollow shaft 9 so as to then enclose the third clutch C3, and has a front end coupled to the carrier PC1.

25 [0068]

The second planetary-gear set G2 and third planetary-gear set G3, the first clutch C1 and second clutch C2, and the first brake B1 and second brake B2 are disposed in a rear space defined between the intermediate wall 8 and the end cover 7 as follows.

30 The second planetary-gear set G2 and third planetary-gear set G3 are disposed on the intermediate shaft 4, wherein the second planetary-gear set G2 is closer to the input shaft 1 than the third

planetary-gear set G3.

The sun gear S2 of the second planetary-gear set G2 and the sun gear S3 of the third planetary-gear set G3 are integrated through the first coupling member M1, and are rotatably supported on the intermediate shaft

5 4.

A clutch drum 15 is arranged to extend radially outward from a roughly middle portion of the hollow shaft 9 and then axially backward to the outer periphery of the second ring gear R2. A clutch pack 16 is arranged comprising an alternate disposition of clutch plates each splined to the inner
10 periphery of the clutch drum 15 and the outer periphery of the ring gear R2. Those members constitute first clutch C1.

[0069]

In order to dispose the second clutch C2 closer to the input shaft 1 than the first clutch C1 located at the outer periphery of the second
15 planetary-gear set G2 as described above, a clutch hub 17 is fixedly mounted to the second sun gear S2 at an outer edge close to the input shaft to extend radially outward. A clutch pack 18 is arranged comprising an alternate disposition of clutch plates each splined to the outer periphery of the clutch hub 17 and the inner periphery of the clutch drum 15.

20 Those members constitute second clutch C2.

A clutch piston 19 of the first clutch C1 and a piston 20 of the second clutch C2 are collectively disposed on the side of the second clutch C2 distant from the first clutch C1 to serves as a double piston wherein the clutch piston 20 slides inside the clutch piston 19, so that the clutch piston
25 20 is fitted in an end wall of the clutch drum 15 facing the second planetary-gear set G2.

The clutch pistons 19, 20 make stroke under the working oil pressure out of respective hydraulic passages 21 (one of which is seen in Figure) formed through the intermediate wall 8 and the hollow shaft 9 so as
30 to allow individual engagement of the first clutch C1 and second clutch C2.

[0070]

The third planetary-gear set G3 comprises a double-sun-gear type

planetary-gear set as described above, wherein the ring gear R3 is smaller in teeth width than the pinion P3 to locate the ring gear R3 meshed with the pinion P3 at an end close to the second planetary-gear set G2, thus allowing shortening of the second coupling member M2 when coupling the ring gear R3 to the carrier PC2 of the second planetary-gear set G2 by the coupling member M2.

A cylindrical coupling member 22, which is disposed to enclose the clutch drum 15 of the first clutch C1 and second clutch C2, has one end connected to the outer periphery of the ring gear R3 and another end connected to the output gear 2.

[0071]

As described above, provided to the carrier PC3 of the third planetary-gear set G3 are center member CM extending radially inward from a side member SM for supporting the third pinion P3 through between the sun gears S3, S4, and outer member OM extending radially outward in the roughly axially middle position of the pinion P3 along an end face of the ring gear R3.

The center member CM is drivingly coupled to the intermediate shaft 4, thereby coupling the carrier PC3 to the clutch drum 11 of the third clutch C3 through the center member CM and the intermediate shaft 4.

A brake hub 23, which is coupled to the outer periphery of the outer member OM, is disposed at the outer periphery of the cylindrical coupling member 22 to extend forward so as to approach the intermediate wall 8.

A brake pack 24 is arranged comprising an alternate disposition of brake plates splined to the outer periphery of a front end of the brake hub 23 and the inner periphery of the transmission casing 3, thereby constituting the first brake B1. The first brake B1 can be engaged as required by a brake piston 25 fitted in the transmission casing 3 at a rear position of the brake pack 24.

[0072]

A brake hub 26 is arranged to conceal a rear end of the brake hub 23, and has a rear-end wall 26a to extend circumferentially inward along the

back of the third planetary-gear set G3. The inner periphery of the brake-hub rear-end wall 26a is coupled to the sun gear S4 of the third planetary-gear set G3.

5 A brake pack 27 is arranged comprising an alternate disposition of brake plates splined to the outer periphery of the brake hub 26 and the inner periphery of the transmission casing 3, thereby constituting the second brake B2. The second brake B2 can be engaged as required by a brake piston 28 fitted in the transmission casing 3 at a rear position of the brake pack 27.

10 With this, the first brake B1 and second brake B2 are disposed at the outer peripheries of the first clutch C1 and second clutch C2, respectively, wherein the first brake B1 is closer to the input shaft 1 (first planetary-gear set G1) than the second brake B2, and wherein the first brake B1 and second brake B2 are closer to the second planetary-gear set G2 than the
15 third planetary-gear set G3.

[0073]

Though omitted in the skeleton views of FIGS. 1 and 4-6, a one-way clutch OWC is arranged between a front end of the brake hub 23 constituting first brake B1 and the transmission casing 3, wherein the first
20 forward speed state is obtained with the first brake B1 released and with one-direction rotation of the third carrier PC3 blocked by the one-way clutch OWC.

At first speed achieved by the one-way clutch OWC, the clutch OWC allows reverse rotation of the third carrier PC3 during engine brake, so that
25 no engine brake is obtained. Thus, upon request of engine brake, the first brake B1 is engaged to block reverse rotation of the third carrier PC3.

A countershaft 29 is separately rotatably arranged in the transmission casing 3 to be parallel to the input shaft 1 and the intermediate shaft 4, and it has a counter gear 30 and a final drive pinion 31 integrated
30 therewith. The counter gear 30 is meshed with the output gear 2, whereas the final drive pinion 31 is meshed with a differential-gear device, not shown, arranged between the automotive driving wheels.

[0074]

With the gear change-speed unit having an actual structure and as shown in FIG. 12, first, since the three planetary-gear sets G1, G2, G3 are arranged in parallel in order of the reduction planetary-gear set G1, single-pinion type planetary-gear set G2, and double sun-gear type planetary-gear set G3 from the side of the input shaft 1, the following operational effect is obtained:

That is, due to higher flexibility of layout of the ring gear R3 about the axial position at the outer periphery of the double sun-gear type planetary-gear set G3 located at the rear end the most distant from the input shaft 1, the ring gear R3 can be positioned closer to the input shaft 1 as shown in FIG. 12 to mesh with the pinion P3, and

the member for ensuring coupling between the carrier PC3 of the double sun-gear type planetary-gear set G3 and the first brake B1 (brake hub 23) for fixing the carrier includes outer member OM which extends radially outward from the carrier PC3 in the roughly axially middle position of the pinion P3, and more specifically, along the end face of the ring gear R3 displaced in the direction of the single-pinion type planetary-gear set G2 (forward) and meshed with the pinion P3,

so that the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3 can radially be narrowed as shown in FIG. 12 without interference from the ring gear R3 and the outer member OM.

As a result, when mounted in horizontal disposition in an automotive engine room, the gear change-speed unit as shown in FIG. 12 can be reduced in the outer periphery of the end portion (end cover 7) distant from the input shaft 1 so as not to interfere with vehicle-body members protruding in the engine room, providing enhanced vehicle mountability in addition to the advantages as described with reference to FIGS. 1 and 4-6.

Moreover, due to a large space which the above arrangement of the ring gear R3 and the outer member OM provides in the vicinity of the outer periphery of the rear end of the third planetary-gear set G3, the end wall

26a of the brake hub 26 connecting the fourth sun gear S4 and the second brake B2 for fixing thereof can be bent in such a way as to enter the space, resulting in sure radial narrowing of the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3 as shown in FIG. 12.

5 [0075]

Further, the first clutch C1 and second clutch C2 of the three clutches C1, C2, C3, which serve to distribute output rotation from the reduction planetary-gear set G1 to the change-speed planetary-gear set comprising double sun-gear type planetary-gear set G2 and single-pinion type planetary-gear set G3, are disposed closer to the single-pinion type planetary-gear set G2 than the double sun-gear type planetary-gear set G3, ,

10

so that the clutches C1, C2 are disposed closer to the input shaft 1, allowing enlargement of a possible area of radially narrowing the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3 regardless of presence of the first clutch C1 and second clutch C2, resulting in further noticeable achievement of the operational effect about vehicle mountability of the gear change-speed unit in FIG. 12 when horizontally disposed in the engine room.

15 20

Still further, the above disposition of the first and second clutches C1, C2 allows those clutches to approach the reduction planetary-gear set G1, reducing the length of the members (hollow shaft 9 and clutch drum 15) for ensuring coupling between the first and second clutches C1, C2 to the reduction planetary-gear set G1, resulting also in achievement of a reduction in length, size, and weight and a simplification of the coupling members.

25

Furthermore, parallel arrangement of the first and second clutches C1, C2 allows double-piston structure of their actuating pistons 19, 20 as described above, leading not only to axial downsizing thereof, but also to reduction in number of parts due to common use of return springs and

30

centrifugal-pressure cancel chambers of the pistons 19, 20 and further to downsizing and cost reduction.

[0076]

In addition to the above disposition of the first and second clutches
5 C1, C2, the pistons 19, 20 of the clutches are disposed on the side of the single-pinion type planetary-gear set G2 distant from the double sun-gear type planetary-gear set G3,

so that the pistons 19, 20 of the first and second clutches C1, C2 are absent at the outer periphery of the double sun-gear type planetary-gear set
10 G3, allowing radial narrowing of the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3, resulting in achievement of the operational effect of enhancing the vehicle mountability of the gear change-speed unit when horizontally disposed in the engine room.

15 [0077]

Furthermore, the third clutch C3 for directly providing rotation of the input shaft 1 to the change-speed planetary-gear set (carrier PC3 of the double sun-gear type planetary-gear set G3 in FIG. 12) comprising double sun-gear type planetary-gear set G3 and single-pinion
20 type planetary-gear set G2 is disposed at the outer periphery of the reduction planetary-gear set G1,

so that the third clutch C3 is located closer to the input shaft 1 than the first and second clutches C1, C2, allowing sure achievement of radial narrowing of the transmission casing (end cover 7) in the vicinity of the
25 outer periphery of the rear end of the double sun-gear type planetary-gear set G3, resulting in sure achievement of the operational effect of enhancing the horizontal-disposition vehicle mountability of the gear change-speed unit. Moreover, this contributes not only to easy routing of the hydraulic passages 21, 14 for the three clutches C1, C2, C3 and reduction in length
30 difference therebetween, but also to improved controllability of the clutches and uniform response for shift in which the clutches are involved.

[0078]

Further, the clutch piston 13 of the third clutch C3 is disposed on the side of the reduction planetary-gear set G1 close to the single-pinion type planetary-gear set G2,

5 so that the clutch piston 13 of the third clutch C3 is disposed opposite and adjacent to the clutch pistons 19, 20 of the first and second clutches C1, C2, resulting in surer achievement of the above operational effect of contributing not only to easy routing of the hydraulic passages 14, 21 for the three clutches C1, C2, C3 and reduction in length difference therebetween, but also to improved controllability of the clutches and
10 uniform response for shift in which the clutches are involved.

[0079]

Still further, the hydraulic passage 21 for the first and second clutches C1, C2 are formed through the intermediate wall 8 as output-gear support wall provided to the transmission casing 3 between the reduction
15 planetary-gear set G1 and single-pinion type planetary-gear set G2,

so that it is possible not only to reduce the length of the hydraulic passage 21 to be arranged between the shift-control control-valve body (not shown) mounted to the transmission casing 3 in any given circumferential position and the first and second clutches C1, C2, but
20 also to roughly equalize the lengths of the hydraulic passages, resulting in uniform response for shift in which the clutches are involved.

Moreover, as for the hydraulic passage 21 for the first and second clutches C1, C2, which needs to lead relatively high pressure due to large transfer torque, it is formed through the intermediate wall 8 having relatively
25 large thickness for supporting the output gear, so that the hydraulic passage 21 can directly be formed through the intermediate wall 8 without any need of a separate and distinct reinforcing sleeve and the like, producing no increase in number of parts and cost.

[0080]

30 Furthermore, the first and second brakes B1, B2 for fixing the rotary members (carrier PC3 and sun gear S4 of the double sun-gear type planetary-gear set G3 in FIG. 12) of the change-speed planetary-gear set

comprising double sun-gear type planetary-gear set G2 and single-pinion type planetary-gear set G3 are disposed closer to the single-pinion type planetary-gear set G2 than the double sun-gear type planetary-gear set G3,

so that even if the two brakes B1, B2 are arranged in association
 5 with the change-speed planetary-gear set comprising double sun-gear type planetary-gear set G2 and single-pinion type planetary-gear set G3, they are disposed closer to the input shaft 1, having no blocking of the above operational effect of radially narrowing the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double
 10 sun-gear type planetary-gear set G3 to enhance the parallel-disposition vehicle mountability of the gear change-speed unit. Moreover, this can roughly equalize the lengths of the hydraulic passages for the brakes B1, B2, resulting in uniform response for shift in which the brakes are involved.
 [0081]

15 Further, the first and second brakes B1, B2 are disposed at the outer periphery of the first and second clutches C1, C2,

so that the coupling members OM, 26a for ensuring coupling between the rotary members (third carrier PC3 and sun gear S4) to be fixed by the brakes B1, B2 and the brakes B1, B2 can be disposed on
 20 the side (rear side) of the double sun-gear type planetary-gear set G3 distant from the single-pinion type planetary-gear set G2, facilitating provision of a mounting space of the coupling members OM, 26a.
 [0082]

The first brake B1 for fixing the carrier PC3 of the double sun-gear
 25 type planetary-gear set G3 is disposed closer to the reduction planetary-gear set G1 than the second brake B2 for fixing the sun gear S4 of the double sun-gear type planetary-gear set G3 distant from the single-pinion type planetary-gear set G2,

so that when extending to the double sun-gear type planetary-gear
 30 set G3 distant from the input shaft 1 the coupling member OM for ensuring coupling between the carrier PC3 of the double sun-gear type planetary-gear set G3 to be fixed by the first brake B1 and the first brake B1,

and the coupling member 26a for ensuring coupling between the sun gear S4 of the double sun-gear type planetary-gear set G3 distant from the single-pinion type planetary-gear set G2 to be fixed by the second brake B2 and the second brake B2, routing of the coupling members OM, 26a is
5 carried out easily in association with the positions of the carrier PC3 and the sun gear S4, and a reduction in length of the coupling members OM, 26a provides great advantages in cost, rigidity, and space efficiency.

[0083]

The hydraulic passage 21 for the first clutch C1 and second clutch
10 C2 is formed through the intermediate wall 8 for supporting the output gear, whereas the hydraulic passage 14 for the third clutch C3 is formed through the pump cover 6, so that all the hydraulic passages are concentratedly arranged at the front of the transmission casing 3 which is advantageous for passing of the hydraulic pressure out of the control-valve body, allowing
15 removal of waste of the shift control circuit.

[0084]

(Second Mode)

FIG. 13 shows gear change-speed unit for an automatic transmission according to another mode of the present invention, wherein
20 like reference numerals designate like parts in FIG. 1.

With the gear change-speed unit (refer to as reduction double-pinion type) for an automatic transmission according to the present mode, first planetary-gear set as reduction gear G1, second planetary-gear set G2 of the single-pinion type, and third planetary-gear set G3 of the double
25 sun-gear type are arranged coaxially in order from a left end close to the input part Input (input shaft 1) in FIG. 13.

The second planetary-gear set G2 and third planetary-gear set G3, which are the same as those described in connection with FIG. 1, constitute Ishimaru-type planetary-gear train (change-speed planetary-gear set).

30 On the other hand, the first planetary-gear set G1 or reduction planetary-gear set comprises, in place of the single-pinion type as described above in connection with FIG. 1, a double-pinion type planetary-gear set

including first sun gear S1, first ring gear R1, and first carrier PC1 for rotatably supporting two first pinions P1a, P1b meshed with the gears S1, R1, respectively, and meshed with each other.

[0085]

5 For this reason, in the present mode, the input shaft 1 is coupled to the first carrier PC1 to receive engine rotation, and the first sun gear S1 is coupled and fixed to the transmission casing 3 at all times. The first ring gear R1 can be coupled as required to the second ring gear R2 through the first clutch C1, and to the second sun gear S2 through the second clutch
10 C2.

Since the other features are the same as those as described above in connection with the FIG. 1, they are illustrated simply while giving the same reference numerals, and a duplicate explanation is avoided.

[0086]

15 By engaging (indicated by circular sign) or releasing (no circular sign) the clutches C1, C2, C3 and the brakes B1, B2 in the combination shown in FIG. 2, the gear change-speed unit according to the present mode having the above structure can select corresponding speed (first to sixth forward speeds and reverse).

20 Shift operation of the gear change-speed unit will be described hereinafter in accordance with FIGS. 2 and 14-17.

FIG. 14 is an alignment chart showing the rotating state of the rotary members of the gear change-speed unit at respective speeds, and FIGS. 15-17 are explanatory views showing torque transfer paths of the gear
25 change-speed unit at respective speeds.

In FIG. 14, the boldest line is alignment chart of the first planetary-gear set G1, and the bold line is alignment chart of the change-speed planetary-gear train (Ishimaru-type planetary-gear train) comprising second planetary-gear set G2 and third planetary-gear set G3.

30 In FIGS. 15-17, the bold line shows a torque-transfer path of the clutches, brakes, and members, and the hatching shows gears for carrying out torque transfer.

[0087]

(First Speed)

As shown in FIG. 2, first speed is obtained by engagement of the first clutch C1 and the first brake B1.

5 At first speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2.

10 On the other hand, in the third planetary-gear set G3, since engagement of the first brake B1 causes fixing of the third carrier PC3 to the casing, rotation of the third sun gear S3 is reverse-direction reduced rotation with respect to output rotation from the third ring gear R3. And rotation of the third sun gear S3 is transferred to the second sun gear S2 of the second planetary-gear set G2 through the first coupling member M1.

[0088]

15 Thus, the second planetary-gear set G2 receives normal-direction reduced rotation through the second ring gear R2 and reverse-direction reduced rotation through the second sun gear S2. Rotation obtained by further decreasing reduced rotation from the second ring gear R2 is provided to the output gear 2 from the second carrier PC2 through the second coupling member M2.

20 That is, as shown in the alignment chart of FIG. 14, first speed is defined by the line connecting an engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second ring gear R2 and an engagement point of the first
25 brake B1 where rotation of the third carrier PC3 is stopped. And rotation input from the input shaft 1 is reduced and provided through the output gear 2.

[0089]

30 A torque-transfer path at first speed is as shown in FIG. 15(a). Thus, torque acts on the first clutch C1, first brake B1, and members shown by the bold line and the first planetary-gear set G1, second planetary-gear set G2, and third planetary-gear set G3 (except the fourth sun gear S4)

shown by the hatching.

Specifically, at first speed, the first planetary-gear set G1 and the second planetary-gear set G2 and third planetary-gear set G3 constituting the Ishimaru-type planetary-gear train are involved in torque transfer.

5 [0090]

(Second Speed)

As shown in FIG. 2, second speed can be obtained by changing of releasing the first brake B1 as engaged at first speed and engaging the second brake B2, and therefore, by engagement of the first clutch C1 and
10 the second brake B2.

At second speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2.

On the other hand, in the third planetary-gear set G3, engagement
15 of the second brake B2 causes fixing of the fourth sun gear S4 to the casing, thus achieving fixing of the third sun gear S3 coupled by the third pinion P3. And the second sun gear S2 coupled to the third sun gear S3 through the first coupling member M1 is fixed to the casing.

[0091]

20 Thus, the second planetary-gear set G2 receives normal-direction reduced rotation through the second ring gear R2, and has the second sun gear S2 fixed. Rotation obtained by further decreasing reduced rotation from the second ring gear R2 is provided to the output gear 2 from the second carrier PC2 through the second coupling member M2.

25 That is, as shown in the alignment chart of FIG. 14, second speed is defined by the line connecting the engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second ring gear R2 and an engagement point of the second brake B2 where rotation of the fourth sun gear S4 is stopped. And rotation
30 input from the input shaft 1 is reduced (but higher than first speed) and provided through the output gear 2.

[0092]

A torque-transfer path at second speed is as shown in FIG. 15(b). Thus, torque acts on the first clutch C1, second brake B1, and members shown by the bold line and the first planetary-gear set G1 and second planetary-gear set G2 shown by the hatching.

5 With the third planetary-gear set G3, the non-fixed third pinion P3, which revolves simply around the fixed sun gears S3, S4 in accordance with output rotation of the third ring gear R3, is not involved in torque transfer though it functions as a rotary member.

[0093]

10 (Third Speed)

As shown in FIG. 2, third speed can be obtained by changing of releasing the second brake B2 as engaged at second speed and engaging the second clutch C2, and therefore, by engagement of the first clutch C1 and second clutch C2.

15 At third speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2. Simultaneously, engagement of the second clutch C2 causes input of this reduced rotation to the second sun gear S2 of the second planetary-gear set G2.

20 Thus, the second planetary-gear set G2 receives the same reduced rotation through the second ring gear R2 and the second sun gear S2, so that reduced rotation (which is the same as that of the first planetary-gear set G1) is provided from the second carrier PC2 which rotates together with the gears R2, S2 to the output gear 2 through the second coupling member M2.

25 [0094]

That is, as shown in the alignment chart of FIG. 14, third speed is defined by the line connecting the engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input
30 rotation to the second ring gear R2 and an engagement point of the second clutch C2 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the second sun gear S2. And rotation input from

the input shaft 1 is reduced (= reduction ratio of the first planetary-gear set G1) and provided through the output gear 2.

A torque-transfer path at third speed is as shown in FIG. 15(c). Thus, torque acts on the first clutch C1 and second clutch C1 and members shown by the bold line and the first planetary-gear set G1 and second planetary-gear set G2 shown by the hatching. That is, the third planetary-gear set G3 is not involved in torque transfer.

[0095]

(Fourth Speed)

As shown in FIG. 2, fourth speed is obtained by changing of releasing the second clutch C2 as engaged at third speed and engaging the third clutch C3, and therefore, by engagement of the first clutch C1 and third clutch C3.

At fourth speed, in the second planetary-gear set G2, engagement of the first clutch C1 causes input of reduced rotation from the first planetary-gear set G1 to the second ring gear R2.

On the other hand, in the third planetary-gear set G3, engagement of the third clutch C3 causes input of input rotation from the input shaft 1 to the third carrier PC3 through the center member CM. As a result, rotation of the third sun gear S3 is increased with respect to that of the third ring gear R3, which is transferred to the second sun gear S2 through the first coupling member M1.

[0096]

Thus, the second planetary-gear set G2 receives reduced rotation through the second ring gear R2 and increased rotation through the second sun gear S2. Rotation (but lower than input rotation) obtained by increasing reduced rotation from the second ring gear R2 is provided to the output gear 2 from the second carrier PC2 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 14, fourth speed is defined by the line connecting the engagement point of the first clutch C1 where reduced rotation from the first planetary-gear set G1 is used as input

rotation to the second ring gear R2 and an engagement point of the third clutch C3 where rotation of the third carrier PC3 is used as input rotation. And rotation input from the input shaft 1 is slightly reduced and provided through the output gear 2.

5 A torque-transfer path at fourth speed is as shown in FIG. 16(a). Thus, torque acts on the first clutch C1, third clutch C3, and members shown by the bold line and the first planetary-gear set G1, second planetary-gear set G2, and third planetary-gear set G3 (except the fourth sun gear S4) shown by the hatching.

10 [0097]

(Fifth Speed)

As shown in FIG. 2, fifth speed is obtained by changing of releasing the first clutch C1 as engaged at fourth speed and engaging the second clutch C2, and therefore, by engagement of the second clutch C2 and third
15 clutch C3.

At fifth speed, engagement of the second clutch C2 causes input of reduced rotation from the first planetary-gear set G1 to the third sun gear S3 through the second sun gear S2 and the first coupling member M1. Simultaneously, engagement of the third clutch C3 causes input of input
20 rotation from the input shaft 1 to the third carrier PC3 through the center member CM.

[0098]

Thus, the third planetary-gear set G3 receives input rotation at the third carrier PC3 and reduced rotation from the first planetary-gear set G1 at
25 the third sun gear S3. Increased rotation with respect to input rotation is provided to the output gear 2 from the third ring gear R3 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 14, fifth speed is defined by the line connecting the engagement point of the second clutch C2
30 where reduced rotation from the first planetary-gear set G1 is used as input rotation to the third sun gear S3 and the engagement point of the third clutch C3 where rotation of the third carrier PC3 is used as input rotation.

And rotation input from the input shaft 1 is slightly increased and provided through the output gear 2.

A torque-transfer path at fifth speed is as shown in FIG. 16(b). Thus, torque acts on the second clutch C2 and third clutch C3 and members shown by the bold line and the first planetary-gear set G1 and third planetary-gear set G3 (except the fourth sun gear S4) shown by the hatching.

[0099]

(Sixth Speed)

As shown in FIG. 2, sixth speed is obtained by changing of releasing the second clutch C2 as engaged at fifth speed and engaging the second brake B2, and therefore, by engagement of the third clutch C3 and the second brake B2.

At sixth speed, engagement of the third clutch C3 causes input of input rotation from the input shaft 1 to the third carrier PC3 through the center member CM. Moreover, engagement of the second brake B2 causes fixing of the fourth sun gear S4 of the third planetary-gear set G3 to the casing.

[0100]

Thus, the third planetary-gear set G3 receives input rotation at the third carrier PC3, and has the fourth sun gear S4 fixed to the casing. Increased rotation with respect to input rotation is provided to the output gear 2 from the third ring gear R3 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 14, sixth speed is defined by the line connecting the engagement point of the third clutch C3 where rotation of the third carrier PC3 is used as input rotation and the engagement point of the second brake B2 where the fourth sun gear S4 is fixed to the casing. And rotation input from the input shaft 1 is increased and provided through the output gear 2.

A torque-transfer path at sixth speed is as shown in FIG. 16(c). Thus, torque acts on the third clutch C3, second brake B2, and members shown by the bold line and the third planetary-gear set G3 (except the third

sun gear S3) shown by the hatching.

[0101]

(Reverse)

As shown in FIG. 2, reverse speed is obtained by engaging the
5 second clutch C2 and the first brake B1.

At reverse speed, engagement of the second clutch C2 causes input
of reduced rotation from the first planetary-gear set G1 to the third sun gear
S3 through the second sun gear S2 and the first coupling member M1. On
the other hand, engagement of the first brake B1 causes fixing of the third
10 carrier PC3 to the casing.

Thus, the third planetary-gear set G3 receives normal-direction
reduced rotation at the third sun gear S3, and has the third carrier PC3 fixed
to the casing. Reverse-direction reduced rotation is provided from the third
ring gear R3 to the output gear 2 through the second coupling member M2.

That is, as shown in the alignment chart of FIG. 14, reverse speed is
15 defined by the line connecting the engagement point of the second clutch
C2 where reduced rotation from the first planetary-gear set G1 is used
as input rotation to the third sun gear S3 and the engagement point of the
first brake B1 where rotation of the third carrier PC3 is stopped. And
20 rotation input from the input shaft 1 is reduced in the reverse direction and
provided through the output gear 2.

A torque-transfer path at reverse speed is as shown in FIG. 17.
Thus, torque acts on the second clutch C2, first brake B1, and members
shown by the bold line and the first planetary-gear set G1 and third
25 planetary-gear set G3 (except the fourth sun gear S4) shown by the
hatching.

[0102]

With the gear change-speed unit for an automatic transmission
according to the present mode, the following operational effects can be
30 obtained in addition to the aforementioned operational effects in the same
way as in the mode shown in FIGS. 1-6:

(H) The gear change-speed unit is constructed by

the reduction double-pinion type first planetary-gear set G1 having first sun gear S1, first ring gear R1, and first carrier PC1 for supporting two first pinions P1a, P1b meshed with the gears S1, R1,

the single-pinion type second planetary-gear set G2 having second
5 sun gear S2, second ring gear R2, and second carrier PC2 for supporting second pinion P2 meshed with the gears S2, R2,

the double sun-gear type third planetary-gear set G3 having two third sun gear S3 and fourth sun gear S4, third carrier PC3 and center member CM for supporting common third pinion P3 meshed with the sun
10 gears S3, S4, and third ring gear R3 meshed with the third pinion P3,

the input shaft 1 coupled to the first carrier PC1,

the output part Output constructed by the output gear (or output shaft) coupled to the second carrier PC2,

the first coupling member M1 for integrally coupling the second sun
15 gear S2 and the third sun gear S3,

the second coupling member M2 for integrally coupling the second carrier PC3 and the third ring gear R3,

the first clutch C1 for selectively engaging and releasing the first ring gear R1 from the second ring gear R2,

the second clutch C2 for selectively engaging and releasing the first
20 ring gear R1 from the second sun gear S2,

the third clutch C3 for selectively engaging and releasing the input shaft Input from the center member CM,

the first brake B1 for selectively stopping rotation of the third carrier
25 PC3, and

the second brake B2 for selectively stopping rotation of the fourth sun gear S3,

so that at first speed and second speed, ring-gear input can be achieved to the Ishimaru-type planetary-gear set comprising second
30 planetary-gear set G2 and third planetary-gear set G3, resulting in further downsizing of the automatic transmission.

In addition, since no torque circulation occurs at second speed, the

transfer efficiency at second speed is enhanced, resulting in achievement of enhanced fuel consumption.

[0103]

FIG. 18 is an actual block diagram of a gear change-speed unit according to the aforementioned mode. When describing hereinafter in detail this mode, in FIG. 18, the input and output parts are shown in the state reversed left to right with respect to those in skeleton views in FIGS. 13 and 15-17.

The input shaft 1 and intermediate shaft 4 are arranged in the transmission casing 3 in the butt state allowing coaxial relative rotation, the input shaft 1 and intermediate shaft 4 being supported rotatably separately with respect to the transmission casing 3.

A front-end opening of the transmission casing 3 close to the input shaft 1 is concealed by the pump casing comprising pump housing 5 and pump cover 6. The input shaft 1, which is arranged through the pump casing for supporting, has a protruding end drivingly coupled to an engine ENG or power source through a torque converter T/C.

[0104]

A rear end of the intermediate shaft 4 distant from the input shaft 1 is rotatably supported by the end cover 7 at a rear end of the transmission casing 3.

The intermediate wall 8 is arranged in a roughly axially middle position of the transmission casing 3 to rotatably support the output gear 2. A front end of the intermediate shaft 4 is rotatably supported in a center hole of the intermediate wall 8 through the hollow shaft 9.

[0105]

The first planetary-gear set G1 is disposed in a front space defined between the pump casing comprising pump housing 5 and pump cover 6 and the intermediate wall 8. The third clutch C3 is disposed in such a way as to enclose the first planetary-gear set G1.

The first planetary-gear set G1 has sun gear S1 fixedly mounted to the pump cover 6 so as to disable rotation at all times, and carrier PC1

couple to the flange 10 extending radially outward from the input shaft 1.

The clutch drum 11 is arranged to extend radially outward from a front end of the intermediate shaft 4 close to the input shaft 1 so as to enclose the ring gear R1 and the clutch hub 32. The clutch pack 12 is
5 arranged comprising an alternate disposition of clutch plates each splined to the inner periphery of the clutch drum 11 and the outer periphery of the clutch hub 32. Those members constitute third clutch C3.

The clutch piston 13 of the third clutch C3 is fitted in an end wall of the clutch drum 11 facing the first planetary-gear set G1, and makes stroke
10 under the working oil pressure out of the hydraulic passage 14 formed through the pump cover 6, the input shaft 1, and the intermediate shaft 4 so as to allow engagement of the third clutch C3.

The drum-shaped coupling member 9a is arranged to extend radially outward from a front end of the hollow shaft 9 so as to then enclose the
15 third clutch C3, and has a front end coupled to the ring gear R1.
[0106]

The second planetary-gear set G2 and third planetary-gear set G3, the first clutch C1 and second clutch C2, and the first brake B1 and second brake B2 are disposed in a rear space defined between the intermediate
20 wall 8 and the end cover 7 as follows.

The second planetary-gear set G2 and third planetary-gear set G3 are disposed on the intermediate shaft 4, wherein the second planetary-gear set G2 is closer to the input shaft 1 than the third planetary-gear set G3.

25 The sun gear S2 of the second planetary-gear set G2 and the sun gear S3 of the third planetary-gear set G3 are integrated through the first coupling member M1, and are rotatably supported on the intermediate shaft 4.

The clutch drum 15 is arranged to extend radially outward from a
30 roughly middle portion of the hollow shaft 9 and then axially backward to the outer periphery of the second ring gear R2. The clutch pack 16 is arranged comprising an alternate disposition of clutch plates each splined to the inner

periphery of the clutch drum 15 and the outer periphery of the ring gear R2. Those members constitute first clutch C1.

[0107]

5 In order to dispose the second clutch C2 closer to the input shaft 1 than the first clutch C1 located at the outer periphery of the second planetary-gear set G2 as described above, the clutch hub 17 is fixedly mounted to the second sun gear S2 at an outer edge close to the input shaft to extend radially outward. The clutch pack 18 is arranged comprising an alternate disposition of clutch plates each splined to the outer periphery of
10 the clutch hub 17 and the inner periphery of the clutch drum 15. Those members constitute second clutch C2.

The clutch piston 19 of the first clutch C1 and the clutch piston 20 of the second clutch C2 are collectively disposed on the side of the second clutch C2 distant from the first clutch C1 to serve as a double piston
15 wherein the clutch piston 20 slides inside the clutch piston 19, so that the clutch piston 20 is fitted in an end wall of the clutch drum 15 facing the second planetary-gear set G2.

The clutch pistons 19, 20 make stroke under the working oil pressure out of the respective hydraulic passages 21 (one of which is seen
20 in Figure) formed through the intermediate wall 8 and the hollow shaft 9 so as to allow individual engagement of the first clutch C1 and second clutch C2.

[0108]

The third planetary-gear set G3 comprises a double sun-gear type
25 planetary-gear set as described above, wherein the ring gear R3 is smaller in teeth width than the pinion P3 to locate the ring gear R3 meshed with the pinion P3 at an end close to the second planetary-gear set G2, thus allowing shortening of the second coupling member M2 when coupling the ring gear R3 to the carrier PC2 of the second planetary-gear set G2 by the
30 coupling member M2.

The cylindrical coupling member 22, which is disposed to enclose the clutch drum 15 of the first clutch C1 and second clutch C2, has one end

connected to the outer periphery of the second coupling member M2 and another end connected to the output gear 2.

[0109]

In the same way as in the aforementioned mode, provided to the carrier PC3 of the third planetary-gear set G3 are center member CM extending radially inward from the side member SM for supporting the pinion P3 through between the sun gears S3, S4, and outer member OM extending radially outward from the carrier PC3 in the roughly axially middle position of the pinion P3.

The center member CM is drivingly coupled to the intermediate shaft 4, thereby coupling the carrier PC3 to the clutch drum 11 of the third clutch C3 through the center member CM and the intermediate shaft 4.

The brake hub 23, which is coupled to the outer periphery of the outer member OM, is disposed at the outer periphery of the cylindrical coupling member 22 to extend forward so as to approach the intermediate wall 8.

The brake pack 24 is arranged comprising an alternate disposition of brake plates splined to the outer periphery of a front end of the brake hub 23 and the inner periphery of the transmission casing 3, thereby constituting the first brake B1. The first brake B1 can be engaged as required by the brake piston 25 fitted in the intermediate wall 8 in front of the brake pack 24.

[0110]

The brake hub 26 is arranged to conceal a rear end of the brake hub 23, and has rear-end wall 26a to extend circumferentially inward along the back of the third planetary-gear set G3. The inner periphery of the rear-end wall 26a is coupled to the sun gear S4 of the third planetary-gear set G3.

The brake pack 27 is arranged comprising an alternate disposition of brake plates splined to the outer periphery of the brake hub 26 and the inner periphery of the transmission casing 3, thereby constituting the second brake B2. The second brake B2 can be engaged as required by the brake

piston 28 fitted in the transmission casing 3 at a rear position of the brake pack 27.

With this, the first brake B1 is disposed at the outer periphery of the first clutch C1 and second clutch C2, whereas the second brake B2 is disposed at the outer periphery of the third planetary-gear set G3, wherein the first brake B1 is closer to the input shaft 1 (first planetary-gear set G1) than the second brake B2.

[0111]

As is omitted in the skeleton views of FIGS. 13 and 15-17, the one-way clutch OWC is arranged between an axially middle portion of the brake hub 23 constituting the first brake B1 and the transmission casing 3, wherein the first forward speed state is obtained with the first brake B1 released and with one-direction rotation of the third carrier PC3 blocked by the one-way clutch OWC.

At first speed achieved by the one-way clutch OWC, the clutch OWC allows reverse rotation of the third carrier PC3 during engine brake to obtain no engine brake, so that upon request of engine brake, the first brake B1 is engaged to block reverse rotation of the third carrier PC3.

A countershaft similar to the countershaft 29 having counter gear 30 and final drive pinion 31 integrated therewith as shown in FIG. 12 is rotatably arranged in the transmission casing 3, through which output rotation of the gear change-speed unit is, of course, provided to the differential-gear device arranged between the automotive driving wheels.

[0112]

In the same way as the aforementioned mode, with the gear change-speed unit shown in FIG. 18 and having the above actual structure, since the three planetary-gear sets G1, G2, G3 are arranged in parallel in order of reduction planetary-gear set G1, single-pinion type planetary-gear set G2, and double sun-gear type planetary-gear set G3 from the side of the input shaft 1, the following operational effects can be obtained:

That is, due to higher flexibility of layout of the ring gear R3 about the axial position at the outer periphery of the double sun-gear type

planetary-gear set G3 located at the rear end the most distant from the input shaft 1, the ring gear R3 can be positioned closer to the input shaft 1 as shown in FIG. 18 to mesh with the pinion P3,

5 and the member for ensuring coupling between the carrier PC3 of the double sun-gear type planetary-gear set G3 and the first brake B1 (brake hub 23) for fixing the carrier includes outer member OM which extends radially outward from the carrier PC3 in the roughly axially middle position of the pinion P3, more specifically, along the end face of the ring gear R3 displaced in the direction of the second planetary-gear set G2
10 (forward) and meshed with the pinion P3,

so that the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3 can radially be narrowed as shown in FIG. 18 without interference from the ring gear R3 and the outer member OM.

15 As a result, when mounted in horizontal disposition in the automotive engine room, the gear change-speed unit as shown in FIG. 18 can be reduced in the outer periphery of the end portion (end cover 7) distant from the input shaft 1 so as not to interfere with vehicle-body members protruding in the engine room, providing enhanced vehicle
20 mountability in addition to the same advantages as described with reference to FIGS. 1 and 4-6.

Moreover, due to a large space which the above arrangement of the ring gear R3 and the outer member OM provides in the vicinity of the outer periphery of the rear end of the third planetary-gear set G3, the end wall
25 26a of the brake hub 26 connecting the fourth sun gear S4 and the second brake B2 for fixing thereof can be bent in such a way as to enter the space, resulting in sure achievement of radial narrowing of the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3 as shown in FIG. 18.

30 This operational effect comes to the fore by bending the outer member OM in such a way as to conceal the outer periphery of the ring gear R3 as shown in FIG. 18 and then extend along the second coupling member

M2. Thus, even with the second brake B2 disposed at the outer periphery of the double sun-gear type planetary-gear set G3, the transmission casing (end cover 7) can radially be narrowed in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3 as shown in FIG. 18.

[0113]

Further, the first clutch C1 and second clutch C2 of the three clutches C1, C2, C3, which serve to distribute output rotation from the reduction planetary-gear set G1 to the change-speed planetary-gear set comprising double sun-gear type planetary-gear set G2 and single-pinion type planetary-gear set G3, are disposed closer to the single-pinion type planetary-gear set G2 than the double sun-gear type planetary-gear set G3,

so that the clutches C1, C2 are disposed closer to the input shaft 1, allowing enlargement of a possible area of radially narrowing the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3 regardless of presence of the first clutch C1 and second clutch C2, resulting in further noticeable achievement of the operational effect about the vehicle mountability of the gear change-speed unit as shown in FIG. 18 when horizontally disposed in the engine room.

Still further, the above disposition of the first clutch C1 and second clutch C2 allows those clutches to approach the reduction planetary-gear set G1, reducing the length of the members (hollow shaft 9 and clutch drum 15) for ensuring coupling between the first and second clutches C1, C2 and the reduction planetary-gear set G1, resulting in achievement of a reduction in length, size, and weight and a simplification of the coupling members.

Furthermore, parallel arrangement of the first and second clutches C1, C2 allows double-piston structure of the operating pistons 19, 20 as described above, leading not only to axial downsizing thereof, but also to reduction in number of parts due to common use of return springs and centrifugal-pressure cancel chambers of the pistons 19, 20 and thus to downsizing and cost reduction of the transmission.

[0114]

In addition to the above disposition of the first and second clutches C1, C2, the clutch pistons 19, 20 of the clutches are disposed on the side of the single-pinion type planetary-gear set G2 distant from the double
5 sun-gear type planetary-gear set G3,

so that the pistons 19, 20 of the first and second clutches C1, C2 are absent at the outer periphery of the double sun-gear type planetary-gear set G3, allowing radial narrowing of the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type
10 planetary-gear set G3, resulting in noticeable achievement of the operational effect of enhancing the vehicle mountability of the gear change-speed unit when horizontally disposed in the engine room.

[0115]

Furthermore, the third clutch C3 for directly providing rotation of the
15 input shaft 1 to the change-speed planetary-gear set (carrier PC3 of the double sun-gear type planetary-gear set G3 in FIG. 18) comprising double sun-gear type planetary-gear set G2 and single-pinion type planetary-gear set G3 is disposed at the outer periphery of the reduction planetary-gear set G1,

so that the third clutch C3 is located closer to the input shaft 1 than
20 the first and second clutches C1, C2, allowing radial narrowing of the transmission casing (end cover 7) in the vicinity of the outer periphery of the rear end of the double sun-gear type planetary-gear set G3, resulting in sure achievement of the aforementioned operational effect of enhancing the
25 horizontal-disposition vehicle mountability of the gear change-speed unit. Moreover, this contributes not only to easy routing of the hydraulic passages 21, 14 for the three clutches C1, C2, C3 and reduction in length difference therebetween, but also to improved controllability of the clutches and uniform response for shift in which the clutches are involved.

[0116]

Further, the clutch piston 13 of the third clutch C3 is disposed on the side of the reduction planetary-gear set G1 close to the single-pinion type

planetary-gear set G2,

so that the piston 13 of the third clutch C3 is disposed opposite and adjacent to the clutch pistons 19, 20 of the first and second clutches C1, C2, resulting in surer achievement of the above operational effect of contributing not only to easy routing of the hydraulic passages 21, 14 for the three clutches and reduction in length difference therebetween, but also to improved controllability of the clutches and uniform response for shift in which the clutches are involved.

[0117]

The hydraulic passage 21 for the first and second clutches C1, C2 is formed through the intermediate wall 8 as output-gear support wall provided to the transmission casing 3 between the reduction planetary-gear set G1 and single-pinion type planetary-gear set G2,

so that it is possible to reduce the length of the hydraulic passage 21 to be arranged between the shift-control control-valve body mounted to the transmission casing 3 in any given circumferential position and the first and second clutches C1, C2, but also to roughly equalize the lengths of the hydraulic passages, resulting in uniform response for shift in which the clutches are involved.

[0118]

Furthermore, the first brake B1 for fixing the carrier PC3 of the double sun-gear type planetary-gear set G3 is disposed closer to the reduction planetary-gear set G1 than the second brake B2 for fixing the sun gear S4 of the double sun-gear type planetary-gear set G3 distant from the single-pinion type planetary-gear set G2,

so that when extending to the double sun-gear type planetary-gear set G3 distant from the input shaft 1 the coupling member OM for ensuring coupling between the carrier PC3 of the double sun-gear type planetary-gear set G3 to be coupled by the first brake B1 and the first brake B1, and the coupling member 26a for ensuring coupling between the sun gear S4 of the double sun-gear type planetary-gear set G3 distant from the single-pinion type planetary-gear set G2 and to be fixed by the second

brake B2 and the second brake B2, routing of the coupling members OM, 26a is carried out easily in association with the positions of the carrier PC3 and the sun gear S4, and a reduction in length of the coupling members OM, 26a provides great advantages in cost, rigidity, and space efficiency.

5 [0119]

The hydraulic passage 21 for the first clutch C1 and second clutch C2 is formed through the intermediate wall 8 for supporting the output gear, whereas the hydraulic passage 14 for the third clutch C3 is formed through the pump cover 6, so that all the hydraulic passages are concentratedly
10 arranged at the front of the transmission casing 3 which is advantageous for passing of the hydraulic pressure out of the control-valve body, allowing removal of waste of the shift control circuit.

[Brief Description of the Drawings]

[FIG. 1] A skeleton view schematically showing gear change-speed unit for an automatic transmission according to a mode of the present invention.
15

[FIG. 2] An engagement-logic explanatory view illustrating the relationship between engagement of the shift friction elements of the gear change-speed unit and selection of the gear ratio.

[FIG. 3] An alignment chart showing the shift state of the gear
20 change-speed unit at respective speeds.

[FIG. 4] Showing torque-transfer paths in the gear change-speed unit at respective speeds, wherein:

(a) is a skeleton view similar to FIG. 1, showing the torque-transfer path at forward first speed,

25 (b) is a skeleton view similar to FIG. 1, showing the torque-transfer path at forward second speed, and

(c) is a skeleton view similar to FIG. 1, showing the torque-transfer path at forward third speed.

[FIG. 5] Showing torque-transfer paths in the gear change-speed unit at
30 respective speeds, wherein:

(a) is a skeleton view similar to FIG. 1, showing the torque-transfer path at forward fourth speed,

(b) is a skeleton view similar to FIG. 1, showing the torque-transfer path at forward fifth speed,

(c) is a skeleton view similar to FIG. 1, showing the torque-transfer path at forward sixth speed.

5 [FIG. 6] A skeleton view similar to FIG. 1, showing a torque-transfer path in the gear change-speed unit when selecting reverse speed.

[FIG. 7] An explanatory view used for explaining torque circulation occurring at second speed in a gear change-speed unit for an automatic transmission, which comprises Ravigneaux-type compound planetary-gear train.

10 [FIG. 8] Showing torque-transfer paths in Simpson-type planetary-gear train and Ravigneaux-type compound planetary-gear train at first speed, wherein:

(a) is a schematic view showing the torque-transfer path in the Simpson-type planetary-gear train at first speed, and

15 (b) is a schematic view showing the torque-transfer path in the Ravigneaux-type planetary-gear train at first speed.

[FIG. 9] An explanatory view showing a difference in tangential force between carrier input and ring-gear input in a planetary-gear set.

20 [FIG. 10] Drawings for explaining impossible achievement of carrier input when obtaining overdrive speed in the case of constructing the gear change-speed unit using Simpson-type planetary-gear train, and a contrivance of the present invention for achieving this, wherein:

(a) is a schematic view for showing rotary-member shortage of the Simpson-type planetary-gear train, in which the above carrier input causes inability,

25 (b) is schematic view of the Simpson-type planetary-gear train, showing that the carrier input is impossible, and

(c) is a schematic view showing the double sun-gar type planetary-gear set showing a contrivance of the present invention which allows the carrier input.

30 [FIG. 11] A view illustrating a performance comparison between the gear change-speed unit using Ravigneaux-type compound planetary-gear train and the gear change-speed unit using Ishimaru-type planetary-gear train.

[FIG. 12] A development sectional view showing an actual structure of the gear change-speed unit in FIGS. 1-6.

[FIG. 13] A skeleton view schematically showing the gear change-speed unit according to another mode of the present invention,

5 [FIG. 14] An alignment chart showing the shift state of the gear change-speed unit at respective speeds,

[FIG. 15] Showing torque-transfer paths in the gear change-speed unit at respective speeds, wherein:

10 (a) is a skeleton view similar to FIG. 13, showing the torque-transfer path at forward first speed,

(b) is a skeleton view similar to FIG. 13, showing the torque-transfer path at forward second speed, and

(c) is a skeleton view similar to FIG. 13, showing the torque-transfer path at forward third speed.

15 [FIG. 16] Showing torque-transfer paths in the gear change-speed unit at respective speeds, wherein:

(a) is a skeleton view similar to FIG. 13, showing the torque-transfer path at forward fourth speed,

20 (b) is a skeleton view similar to FIG. 13, showing the torque-transfer path at forward fifth speed, and

(c) is a skeleton view similar to FIG. 13, showing the torque-transfer path at forward sixth speed.

[FIG. 17] A skeleton view similar to FIG. 13, showing a torque-transfer path in the gear change-speed unit when selecting reverse speed.

25 [FIG. 18] A development sectional view showing an actual structural view of the gear change-speed unit shown in FIGS. 13-17.

[Description of the Reference Numerals]

G1: first planetary-gear set (reduction planetary-gear set)

30 G2: second planetary-gear set (single-pinon type planetary-gear set)

G3: third planetary-gear set (double sun-gear type planetary-gear set)

	M1:	first coupling member
	M2:	second coupling member
	C1:	first clutch
	C2:	second clutch
5	C3:	third clutch
	B1:	first brake
	B2:	second brake
	Input:	input part
	1:	input shaft
10	Output:	output part
	2:	output gear
	S1:	first sun gear
	R1:	first ring gear
	P1:	first pinion
15	P1a:	first pinion
	P1b:	second pinion
	PC1:	first carrier
	S2:	second sun gear
	R2:	second ring gear
20	P2:	second pinion
	PC2:	second carrier
	S3:	third sun gear
	S4:	fourth sun gear
	P3:	third pinion
25	PC3:	third carrier
	R3:	third ring gear
	CM:	center member
	SM:	side member
	OM:	outer member
30	ECG:	engine (power source)
	T/C:	torque converter
	3:	transmission casing

	4:	intermediate shaft
	5:	pump housing
	6:	pump cover
	7:	end cover
5	8:	intermediate wall (output-gear support wall)
	9:	hollow shaft
	9a:	drum-shaped coupling member
	10:	flange
	11:	clutch drum
10	12:	clutch pack
	13:	clutch piston
	14:	third clutch hydraulic passage
	15:	clutch drum
	16:	clutch pack
15	17:	clutch hub
	18:	clutch pack
	19:	clutch piston
	20:	clutch piston
	21:	first-clutch or second-clutch hydraulic passage
20	22:	cylindrical coupling member
	23:	brake hub
	24:	brake pack
	25:	brake piston
	26:	brake hub
25	26a:	brake-hub rear end wall
	27:	brake pack
	28:	brake piston
	29:	countershaft
	30:	counter gear
30	31:	final drive pinion
	32:	clutch hub

[Document Name] Drawings

[FIG. 2]

- 1: friction elements
- 2: speeds
- 5 3: forward
- 4: first speed
- 5: second speed
- 6: third speed
- 7: fourth speed
- 10 8: fifth speed
- 9: sixth speed
- 10: reverse

[FIG. 3]

- 11: overdrive (OD)
- 15 12: sixth speed
- 13: fifth speed
- 14: third speed
- 15: second speed
- 16: first speed
- 20 17: reverse

[FIG. 4]

- 18: (a) first speed
- 19: (b) second speed
- 20: (c) third speed

25 [FIG. 5]

- 21: (a) fourth speed
- 22: (b) fifth speed
- 23: (c) sixth speed

[FIG. 6]

- 30 24: reverse

[FIG. 7]

- 25: second speed

26: Ravigneaux-type compound planetary-gear train
[FIG. 11]

- | | | |
|----|-----|--|
| | 27: | planetary-gear set |
| | 28: | gear ratio |
| 5 | 29: | gear-to-gear ratio |
| | 30: | forward reverse ratio |
| | 31: | transfer ratio |
| | 32: | engagement-element torque share |
| 10 | 33: | engagement-element increasing number when adopting OWC |
| | 34: | cover ratio |
| | 35: | direct coupling mode |
| | 36: | 7 speed ratio |
| | 37: | first speed |
| 15 | 38: | second speed |
| | 39: | third speed |
| | 40: | fourth speed |
| | 41: | fifth speed |
| | 42: | sixth speed |
| 20 | 43: | reverse |
| | 44: | first speed/second speed |
| | 45: | second speed/third speed |
| | 46: | third speed/fourth speed |
| | 47: | fourth speed/fifth speed |
| 25 | 48: | fifth speed/sixth speed |
| | 49: | reverse/first speed |
| | 50: | first speed |
| | 51: | second speed |
| | 52: | third speed |
| 30 | 53: | fourth speed |
| | 54: | fifth speed |
| | 55: | sixth speed |

	56:	seventh speed
	57:	reverse
	58:	total
	59:	minimum
5	60:	maximum
	61:	sixth speed
	62:	ratio coverage
	63:	ratio coverage
	64:	Ravigneaux type
10	65:	Ishimaru type
	66:	reduction double pinion
	67:	reduction single pinion
	68:	Ravigneaux type
	69:	Ishimaru type
15	70:	reduction double pinion
	71:	reduction single pinion
	72:	none
	73:	possible
	74:	none
20	75:	possible
	76:	none
	77:	possible
	78:	none
	79:	possible
25	80:	none
	81:	possible
	82:	none
	83:	possible
[FIG. 14]		
30	84:	sixth speed
	85:	fifth speed
	86:	third speed

87: second speed

88: fourth speed

89: first speed

90: reverse

5 [FIG. 15]

91: (a) first speed

92: (b) second speed

93: (c) third speed

[FIG. 16]

10 94: (a) fourth speed

95: (b) fifth speed

96: (c) sixth speed

[FIG. 17]

97: reverse

15

[Document Name] Abstract

[Abstract]

[Problem to be Solved] To achieve a structure which actualizes O/D speed while preserving the strength advantage of a Simpson-type planetary-gear train without relying on parallel disposition of input and output parts and allows a reduction in rear-end outer diameter of a transmission casing.

[Solving Means] A reduction planetary-gear set G1, a single-pinon type planetary-gear set G2, and a double sun-gear type planetary-gear set G3 are disposed in parallel in this order from the side of an input shaft. Those

planetary-gear sets, clutches C1-C3, and brakes B1, B2 constitute a gear change-speed unit which allows coaxial disposition of the input shaft 1 and an output gear 2 and actualization of six forward speeds including O/D speed. A ring gear R3 of G3 at the rear end is located close to the input shaft to mesh with a pinion P3. A member OM for ensuring coupling

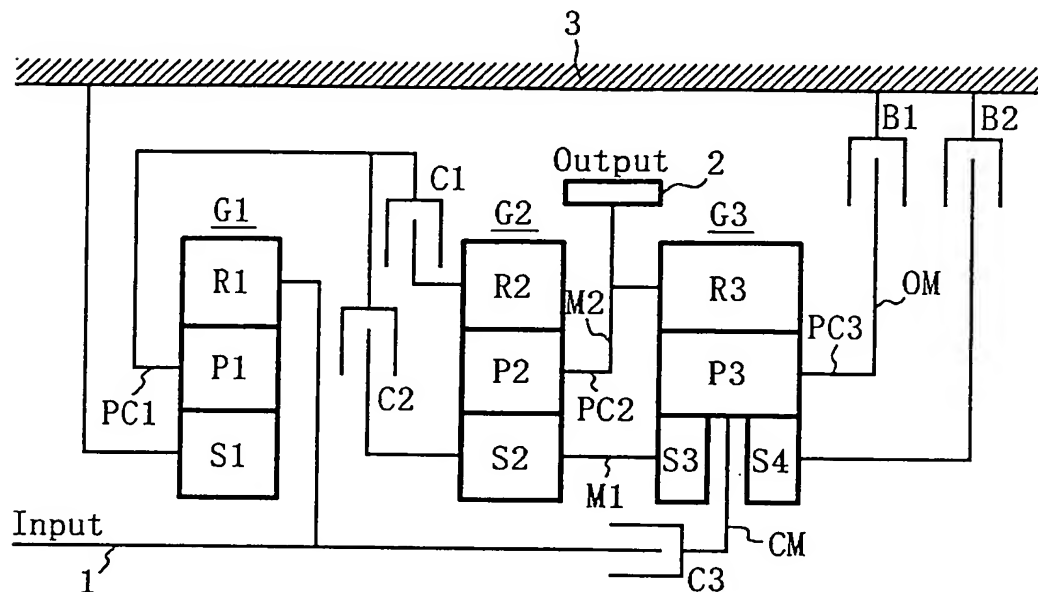
between a carrier PC3 of G3 and a first brake B1 (hub 23) for fixing thereof includes an outer member OM extending radially outward from PC3 in the roughly axially middle position of P3 along the end face of R3. Thus,

a large space is produced in the vicinity of the rear-end outer periphery of G3, and a brake-hub end wall 26a for connection between a sun gear S4 and the second brake B2 can be bent to enter the space, allowing radial narrowing of a transmission casing (end cover 7) in the vicinity of the rear-end outer periphery of G3, resulting in no interference with a vehicle member protruding in an engine room when horizontally disposing the gear change-speed unit.

[Drawing Selected] FIG. 12

【書類名】 図面

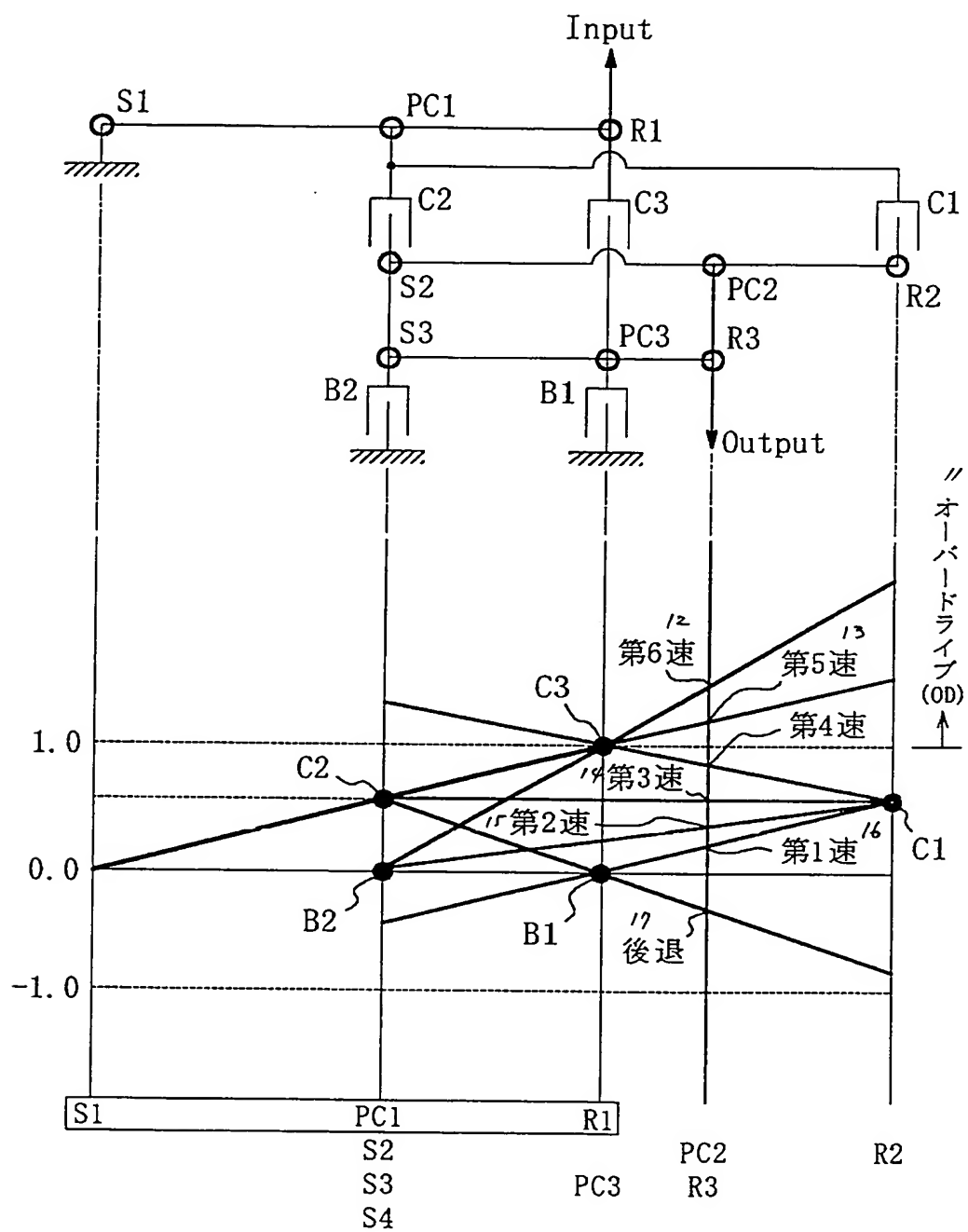
【図1】



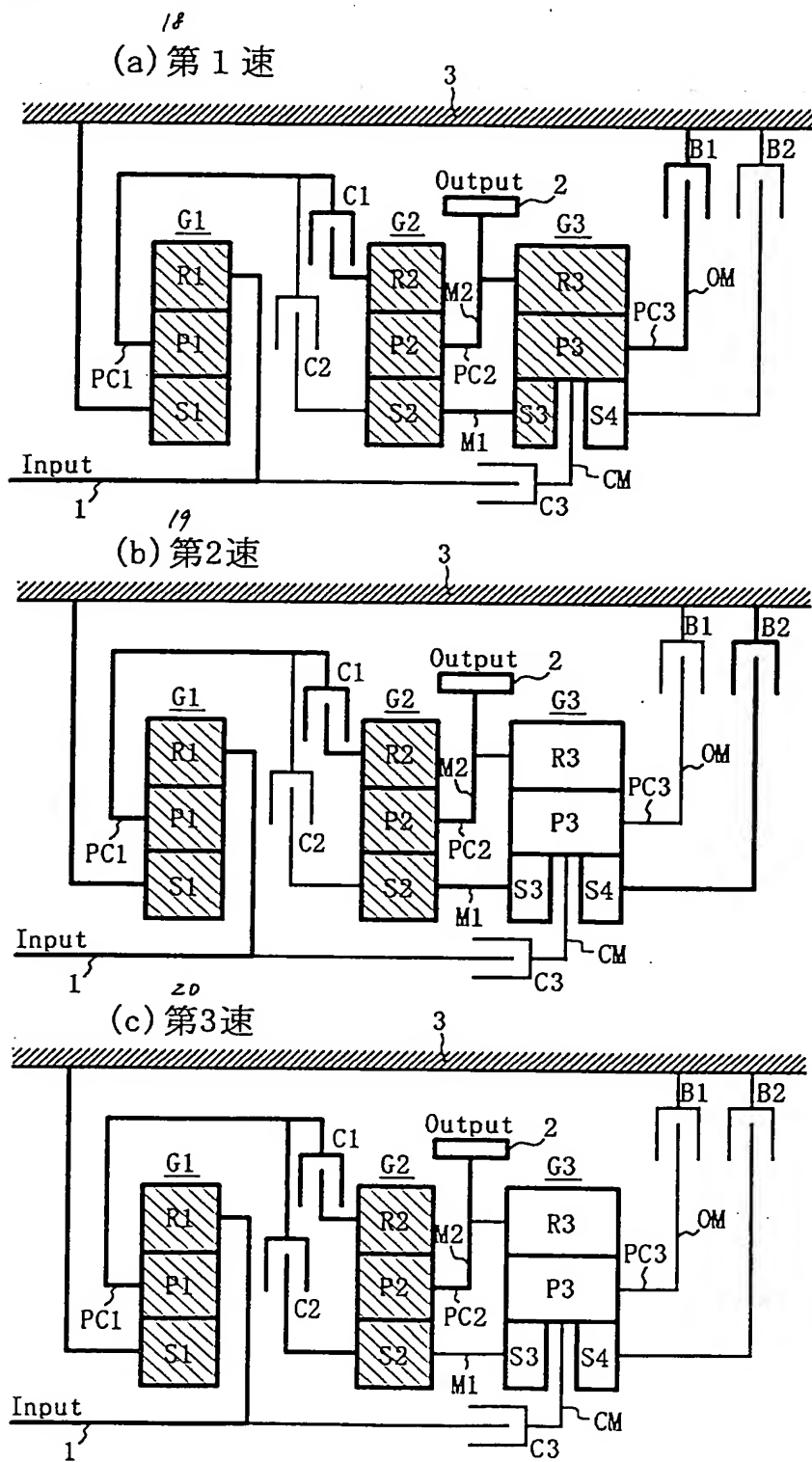
【図2】

摩擦要素		C1	C2	C3	B1	B2	5.5	6.0	6.5	7.0
変速段										
前進	第1速	○			○		4.060	4.260	4.583	4.782
	第2速	○				○	2.192	2.360	2.500	2.773
	第3速	○	○				1.538	1.600	1.677	1.818
	第4速	○		○			1.153	1.164	1.170	1.205
	第5速		○	○			0.891	0.870	0.862	0.824
	第6速			○		○	0.741	0.714	0.714	0.678
後退			○		○		4.396	4.000	4.167	3.828
						α 1	0.350	0.375	0.400	0.450
						α 2	0.350	0.400	0.400	0.475
						α 3	0.425	0.475	0.500	0.525

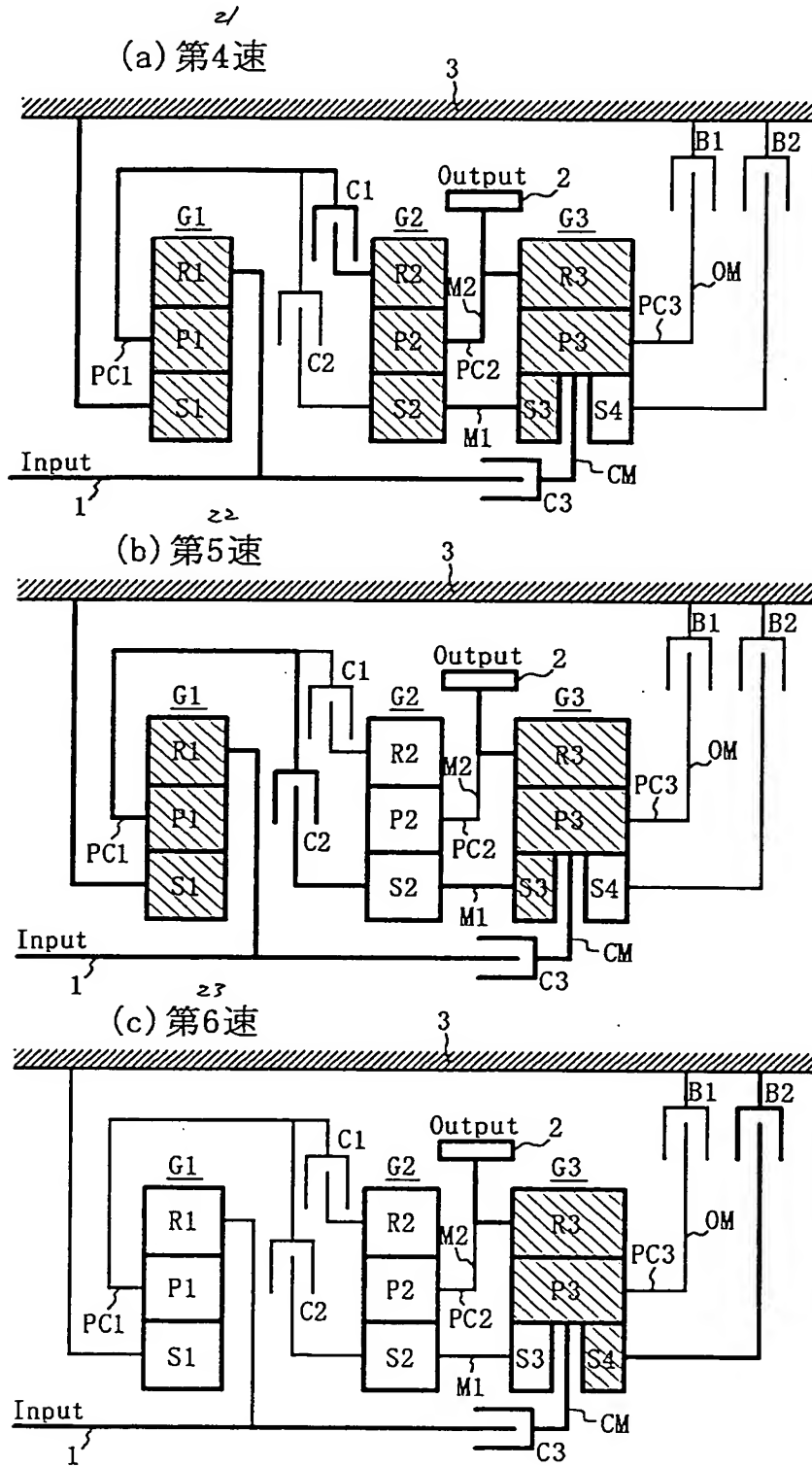
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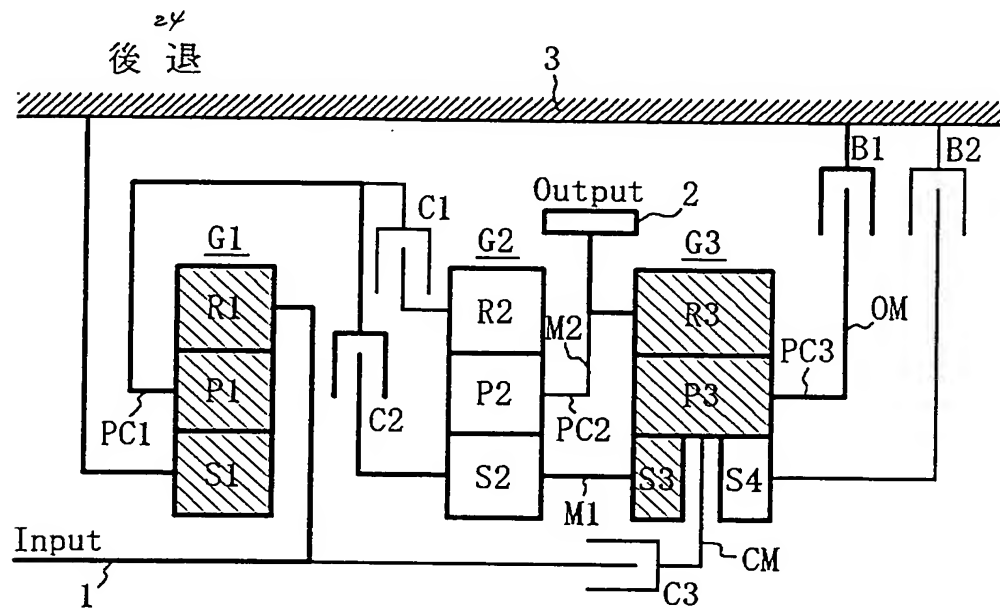
【図4】



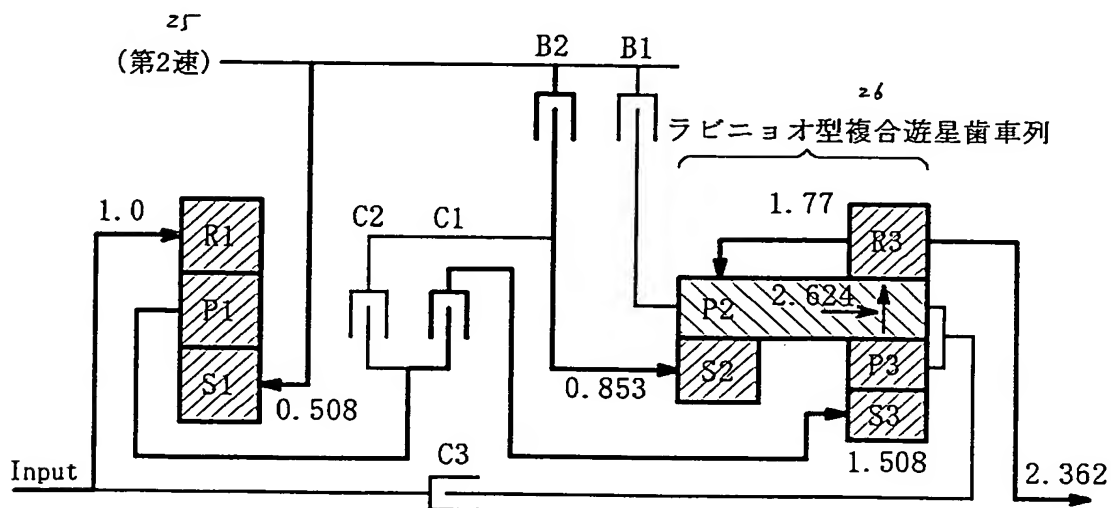
【図5】



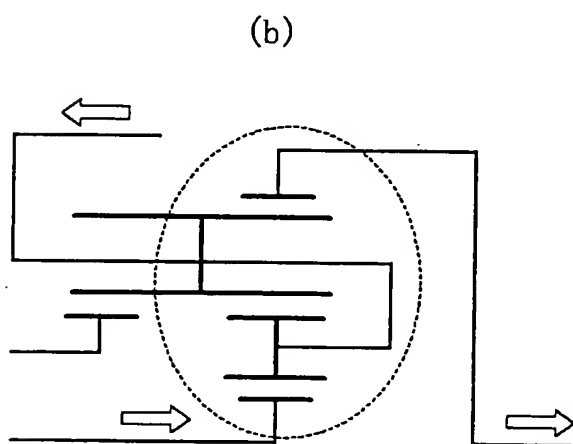
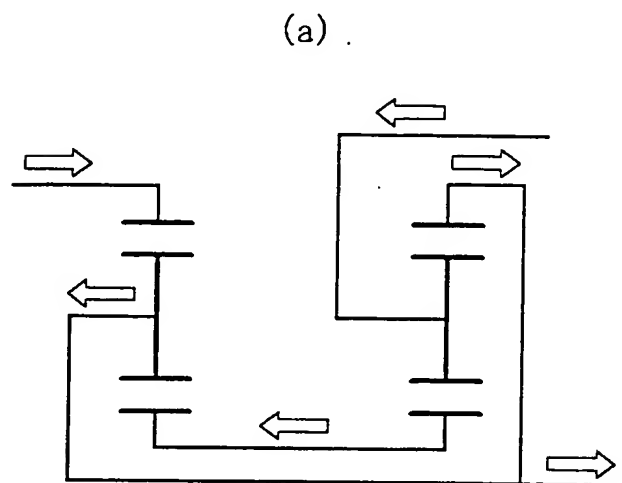
【図6】



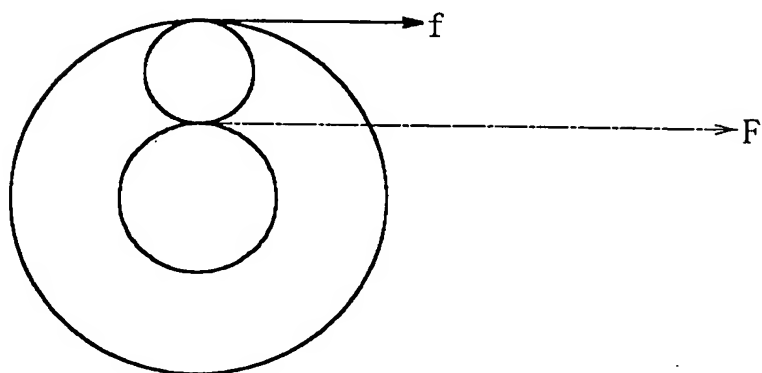
【図7】



【図8】

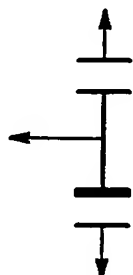


【図9】

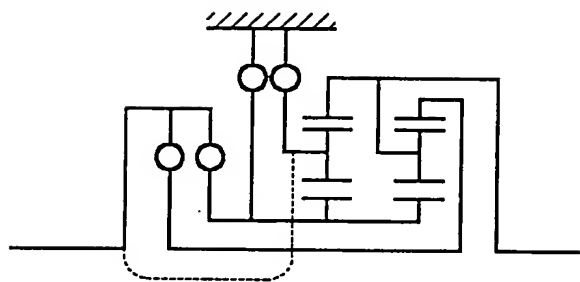


【図10】

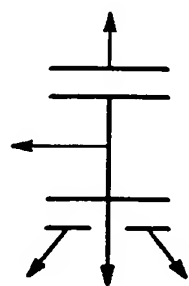
(a)



(b)



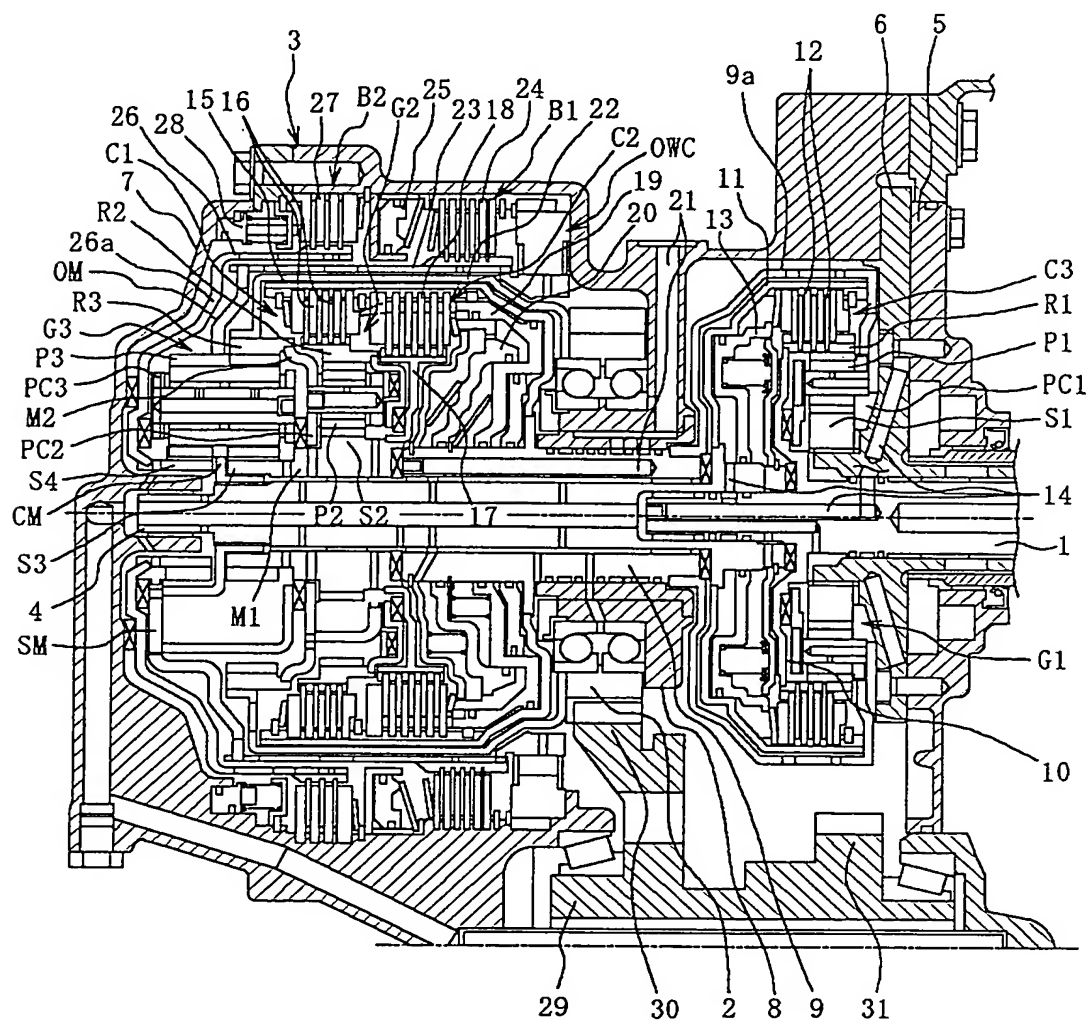
(c)



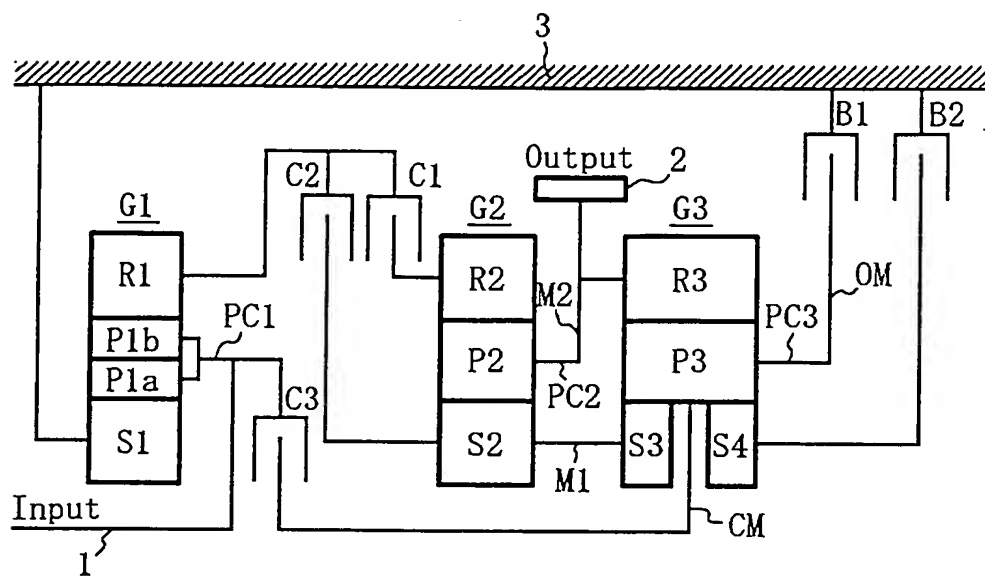
【図11】

		6/ 6 速						
		62 レシオカバレッジ:1			63 レシオカバレッジ:2			
		64 ラビニョオ型	65 イシマル型		68 ラビニョオ型	69 イシマル型		
			66 減速ダブルピニオン	67 減速シングルピニオン		70 減速ダブルピニオン	71 減速シングルピニオン	
27	遊星歯車比	α 1	0.575	0.350	0.550	0.650	0.425	0.625
		α 2	0.375	0.350	0.500	0.475	0.350	0.550
		α 3	0.350	0.500	0.375	0.350	0.500	0.350
28	変 速 比	37 第1速	4.500	4.505	4.392	4.714	5.093	5.072
		38 第2速	2.373	2.308	2.325	2.637	2.609	2.519
		39 第3速	1.575	1.538	1.550	1.650	1.739	1.625
		40 第4速	1.146	1.136	1.148	1.160	1.170	1.141
		41 第5速	0.880	0.891	0.883	0.842	0.872	0.881
		42 第6速	0.727	0.741	0.727	0.678	0.741	0.741
		43 後 退	4.200	4.395	4.133	3.474	4.969	4.634
29	段 間 比	44 第1速/第2速	1.896	1.952	1.889	1.788	1.952	2.013
		45 第2速/第3速	1.507	1.501	1.500	1.598	1.500	1.550
		46 第3速/第4速	1.374	1.354	1.356	1.422	1.488	1.424
		47 第4速/第5速	1.302	1.275	1.294	1.378	1.342	1.295
		48 第5速/第6速	1.210	1.202	1.215	1.242	1.177	1.189
30	前後進比	49 後退/第1速	0.933	0.976	0.941	0.737	0.976	0.914
31	伝達効率	50 第1速	0.968	0.969	0.974	0.968	0.989	0.974
		51 第2速	0.950	0.968	0.972	0.952	0.968	0.972
		52 第3速	0.993	0.988	0.993	0.993	0.988	0.993
		53 第4速	0.982	0.987	0.989	0.983	0.988	0.989
		54 第5速	0.989	0.988	0.989	0.989	0.989	0.990
		55 第6速	0.993	0.993	0.993	0.993	0.993	0.993
		56 第7速						
32	締結要素 トルク分担	57 後 退	0.978	0.973	0.978	0.978	0.973	0.978
		C1	1.575	1.203	1.550	1.650	1.175	1.625
		C2	1.575	1.538	1.550	1.650	1.739	1.625
		C3	1.209	1.538	1.214	1.243	1.739	1.190
		B1	5.775	0.769	5.683	5.124	0.909	6.268
		B2	0.798	5.934	0.775	0.987	6.708	0.894
33	OWC採用時の締結 要素増加個数	f5 合 計	10.932	10.982	10.772	10.654	12.270	11.602
		OWC1	0	0	0	0	0	0
		OWC2	1	1	1	1	1	1
34	レシオカバレッジ	59 最 小	4.81	5.08	4.81	4.81	5.08	4.81
		60 最 大	7.20	9.02	7.80	7.20	9.02	7.80
35	直結モード	72 無 し	74 無 し	76 無 し	78 無 し	80 無 し	82 無 し	
36	7 速 比	73 可 能	75 可 能	77 可 能	79 可 能	81 可 能	83 可 能	

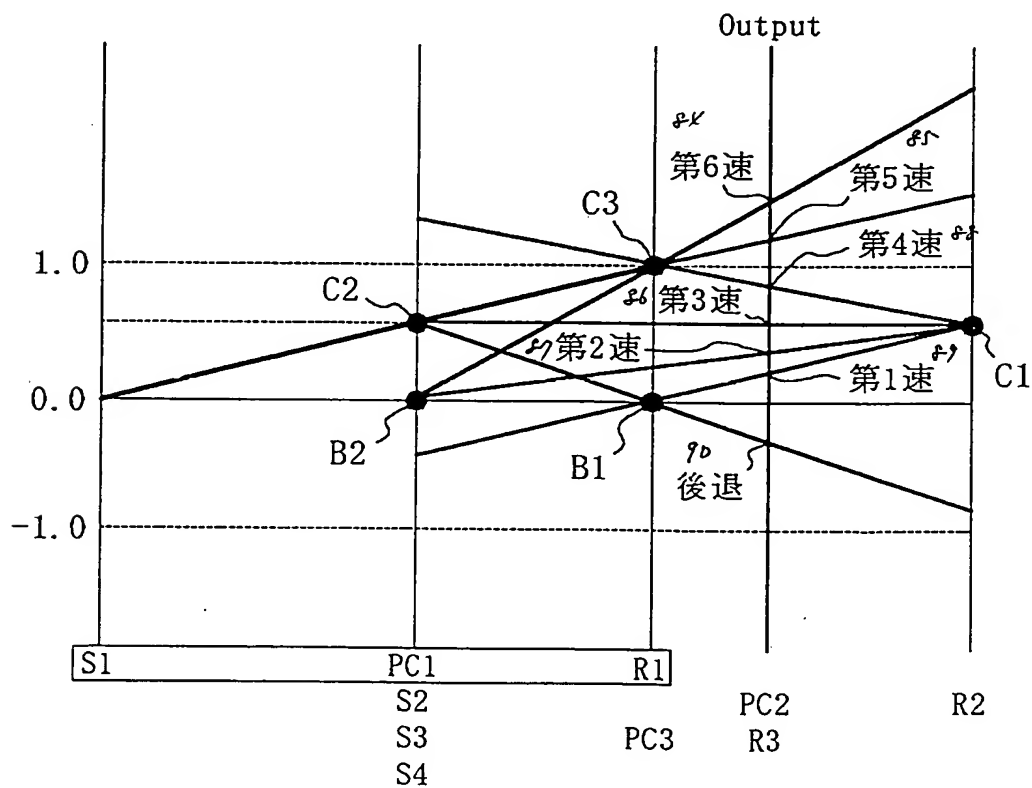
【図12】



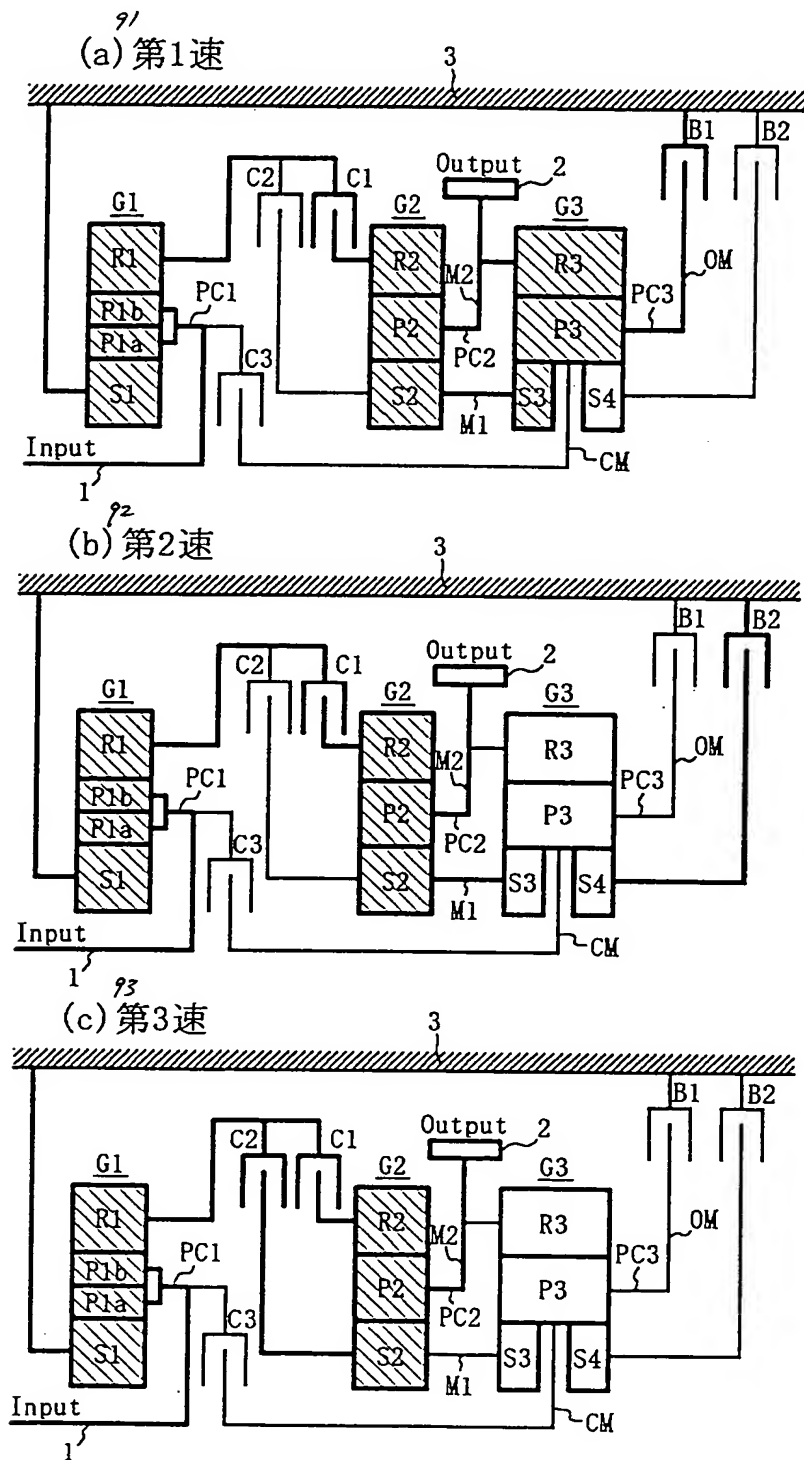
【図13】



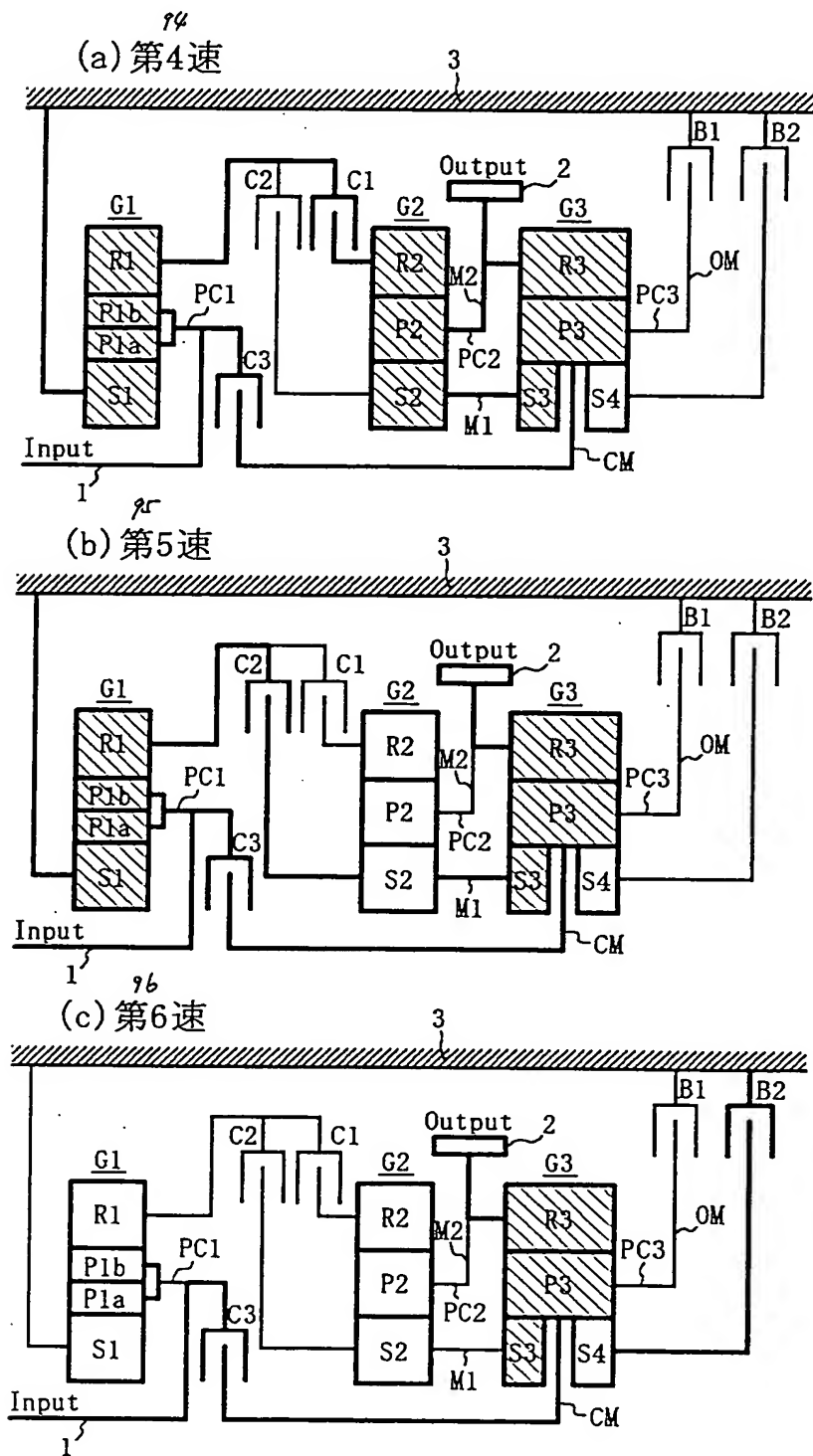
【図14】



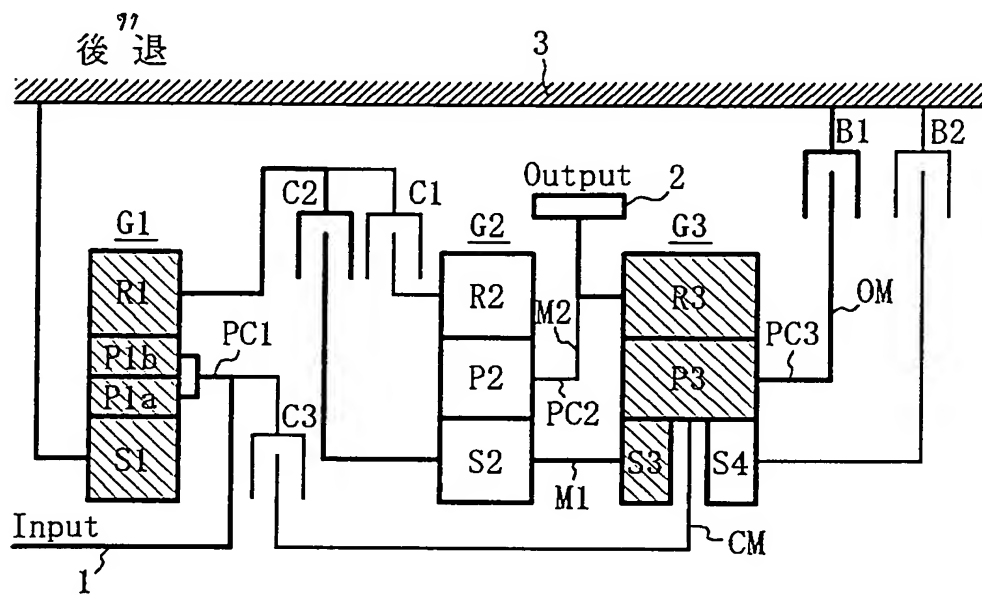
【図15】



【図16】



【図17】



【図18】

